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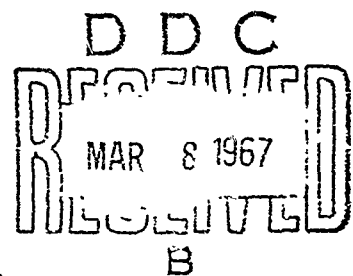
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INVESTIGATION OF A HIGH SPEED
DIRECTLY DRIVEN ENGINE
HYDRAULIC PUMP

F. W. Perian
Aerospace Division
Vickers Incorporated
Division of Sperry Rand Corporation

October 1966

Technical Report AFAPL-TR-66-103



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Air Force Aero Propulsion Laboratory
Research and Technology Division
Air Force Systems Command
Wright-Patterson Air Force Base, Ohio 45433

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FOREWORD

This report was prepared by the Aerospace Division, Vickers Incorporated of the Sperry Rand Corporation, Troy, Michigan, under USAF Contract AF33(615)1603. The contract was conducted under Project No. 8128, Task No. 812807. The Vickers Project No. and Report No. is 8-0108-077C. Administration was under the Air Force Aero Propulsion Laboratory, Research and Technology Division, Air Force Systems Command, Wright-Patterson Air Force Base, with Mr. K. E. Binns acting as Project Engineer.

The program was conducted with Mr. F. W. Perian as principal investigator, under the direction of Mr. M. P. Donnelly, Chief Engineer - Packages and Components.

The project was started with the initiation of the study phase in June of 1964. The test phase of the project was completed in May of 1966. The report was submitted by the author October 1966.

Publication of this report does not constitute Air Force approval of the report's findings, or conclusions. It is published only for the exchange and stimulation of ideas.

PAUL MONTGOMERY, Major USAF
Chief, Propulsion & Power Branch
Aerospace Power Division
Wright-Patterson Air Force Base, Ohio

ABSTRACT

This report covers an exploratory development program to establish design technology for hydraulic pumps capable of being driven directly from aircraft gas turbine propulsion engines. The program was divided into a study phase, design phase, fabrication phase, and a test phase.

After completing the study phase, the orbital type pump was chosen by the Air Force Propulsion Laboratory for design, fabrication, and test. Other pumping concepts were available that would theoretically reach the contract goals. However, it was desired that a unit be developed that would show growth potential beyond 10 gpm and 18,000 rpm.

The .140 cubic inch per revolution orbital pump was designed from an existing .70 cubic inch per revolution unit by using a .585 scale factor. The porting was designed for 10 gpm flow, and the bearings were designed for 1000 hours life. The pump was designed to be compatible with MIL-H-5606 fluid in a Type II system.

The test phase was conducted in accordance with the development test program for the E80108075 orbital piston pump. The tests were to determine pump performance and endurance capabilities. An investigation was also made to determine the horsepower loss for the various rotating parts of the pump.

The orbital pump completed the Examination of Product, Break-in Run, Functional Test, Calibration, Proof Pressure Test, Sampling Test, and Low Temperature Test.

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SECTION I

INTRODUCTION

This program was investigative in nature and was intended to acquire design technology and criteria for developing hydraulic pumps capable of being directly driven from aircraft gas turbines. The program was directed towards obtaining technology for an 18,000 rpm, 10 gpm, 3000 psi pump capable of operating reliably for 1000 hours. This was later changed to a 500 hour requirement. (See page 37)

The program was divided into four major phases; study, design, fabrication and test. These are briefly described below.

The object of the study phase of the program was to determine the most suitable pump concept. The orbital pump was chosen because it showed growth potential beyond the contract requirements.

The design phase of the program was used to review the stress levels, life, material compatibility, fluid velocity, and power losses of the orbital pump concept.

The fabrication phase provided one E80108075 pump and spare parts. The spare parts fabricated and purchased were normally replaceable items. Any items that were considered subject to further development were not ordered as spares.

The test phase of the program was to determine whether the orbital pump was capable of the contract goals of 18,000 rpm, 10 gpm, 1000 hour life. The pump was tested to the requirements of MIL-P-25868, as stated in Exhibit "A" of AF33(615)1603 contract.

The pumping concept was proven through test. Further development of materials and processes is necessary to provide the required reliability and operational life.

Recommendations will be discussed under Section VI.

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SECTION II

STUDY OF PUMP CONCEPTS

To achieve the design goals of the program, the following pumping concepts were considered:

1. Bent Axis Piston Pump
2. Axial (In-Line) Piston Pump
3. Vane Pump
4. Orbital Piston Pump
5. Gear Pump
6. Radial Piston Pump
7. Centrifugal Pump
8. Screw Pump

The comparisons were made utilizing actual test data, analysis calculations and technical experience. The results of these comparisons were tabulated and weighted to arrive at a numerical value so that the proper type pump would be selected.

The variable displacement requirement in the contract eliminated the gear pump and the screw pump. The centrifugal pump was reviewed with the aid of the "ASD Hydraulic Pump Investigation" report under Contract No. AF33(657)10151. The calculated overall efficiency was 25 percent. This eliminated the centrifugal pump from further consideration.

The five types of pumps remaining were broken into categories as follows:

1. Bent Axis Piston Pump
 - (a) Yoke
 - (b) Fixed Angle Variable (FAV)
2. Axial (In-Line) Piston Pump
3. Vane Pump
 - (a) Concentric Ring
 - (b) Fixed Displacement Variable
4. Orbital Piston Pump

5. Radial Piston Pump

At this point, performance figures were generated through calculations and extrapolation of test data. Results of the performance analysis are shown graphically on the bar chart of Table I. Power losses were obtained at both full delivery and at zero delivery. The calculation of the zero delivery losses was considered necessary since most aircraft systems operate at or near zero pump delivery a majority of the time.

1. BENT AXIS PISTON PUMP

There are two methods used to regulate the flow of the bent axis piston pumps. The first is to vary the displacement of the unit by changing the stroke of the pistons. The second is to vary the delivery of the pump by an internal porting arrangement.

(a) Yoke Pump

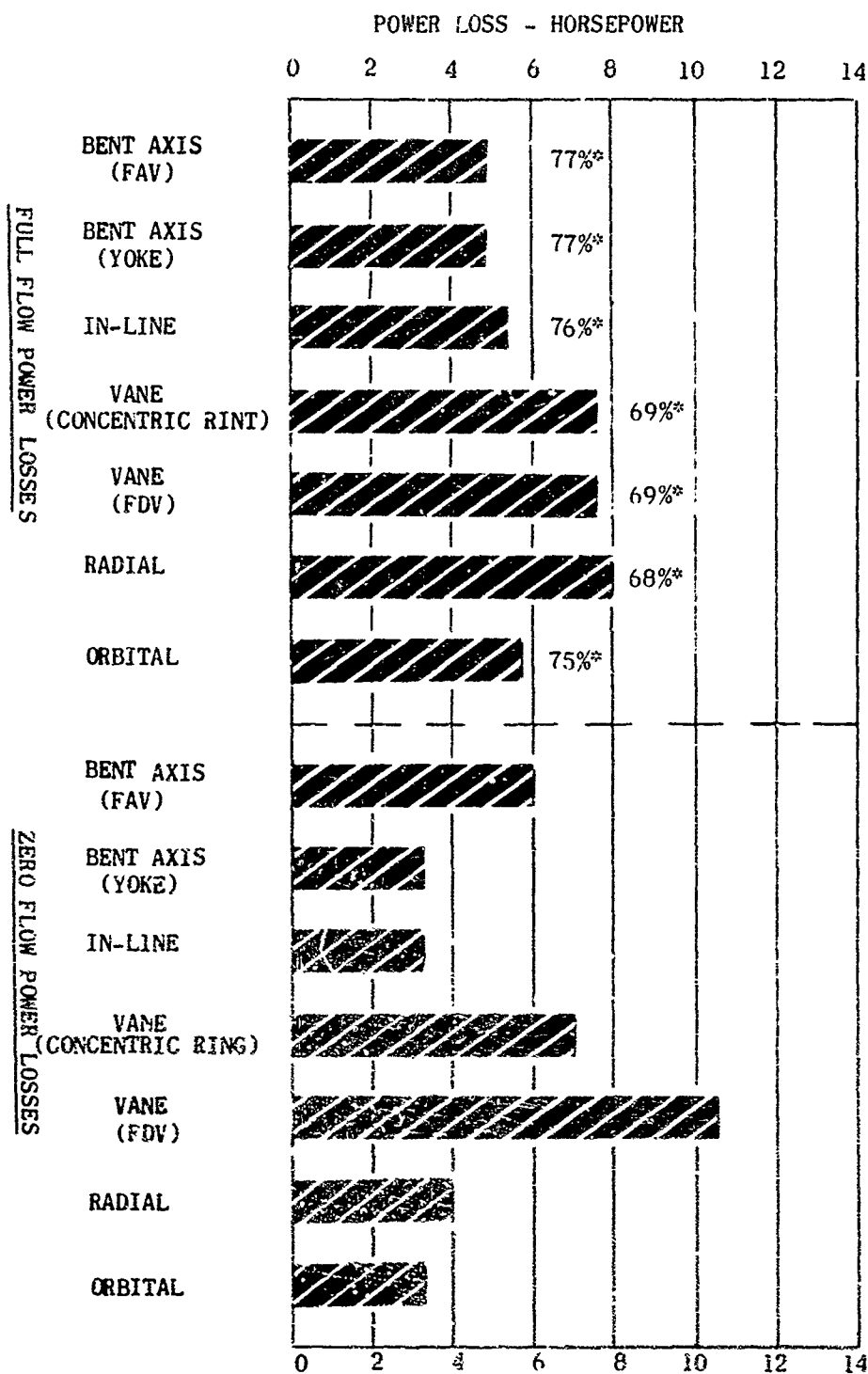
In the yoke pump, the porting for the cylinder block is mounted in a bearing mounted yoke. (Figure 1) The inlet and outlet to the cylinder block are through the yoke pintles. The drive shaft is fixed in the pump housing. Swinging the yoke to one side of the drive shaft center line results in an increasing stroke between the pistons that are attached to the drive shaft and the cylinder block bores. A universal link maintains the correct orientation between the drive shaft and the cylinder block. Yoke position is achieved by a linear actuator controlled by a pressure sensitive control valve.

(b) Fixed Angle Variable (FAV)

The basic pumping element of the FAV pump is the same as the yoke pump. The method of regulating pump delivery is different. (Figure 2) The displacement of the FAV pump is constant at all times; only the pump outlet flow is varied.

The variable delivery is obtained by rotating the rotor. The rotor contains the cylinder block porting and is mounted between the cylinder block and valve plate. Pump delivery is controlled by rotating the valve plate in relation to the fixed pump angle. Rotation of 90° allows all of the pump delivery to be recirculated within the pumping element. A lesser rotation permits only a portion to be recirculated and the rest delivered to the system. The angular rotation is controlled by a rotary actuator torsion spring and pressure sensitive control valve.

TABLE I
COMPARISON OF PUMP TYPES



* Overall Efficiency

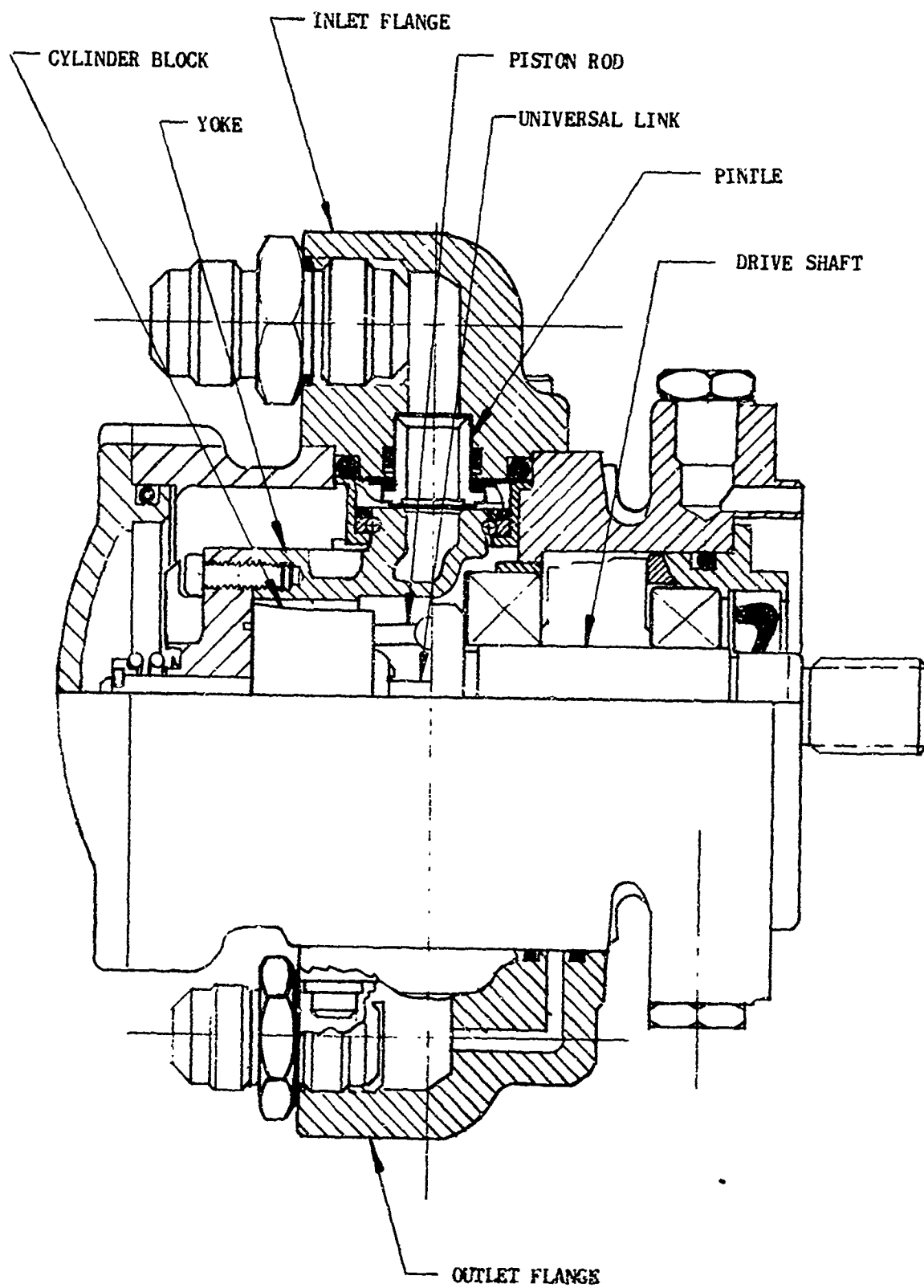


FIGURE 1 - YOKE PUMP

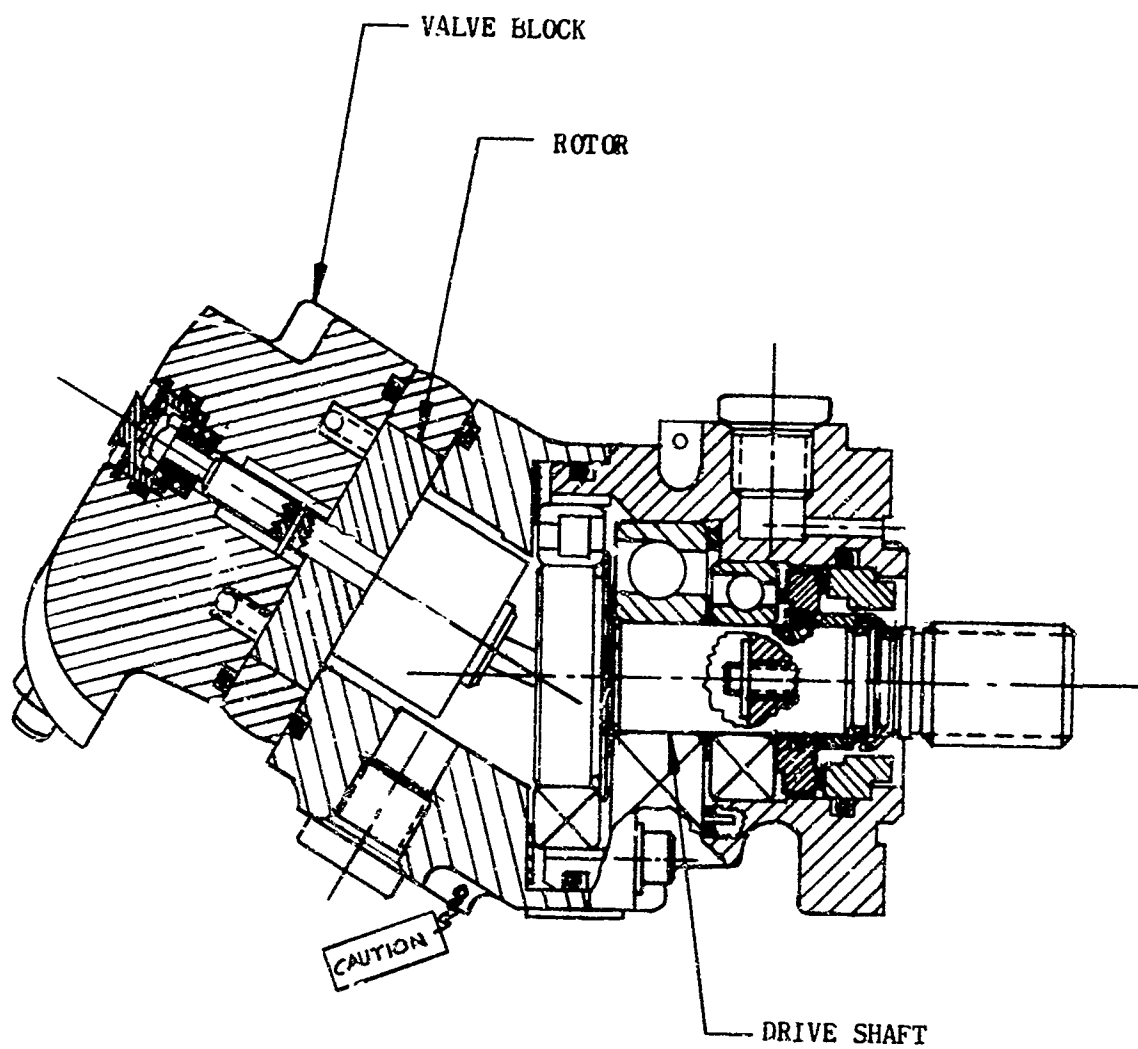


FIGURE 2 - FAV PUMP

Actual test data was used to generate the performance values for Table I for both the yoke and FAV pumps.

2. AXIAL (IN-LINE) PISTON PUMPS

The type of in-line pump considered for this application incorporated a variable angle swash-plate. (Figure 3) In this pump, the swash-plate is mounted on two trunion bearings (not shown in Figure 3). The swash-plate angle, which determines the piston movement, is controlled by a pressure sensitive control valve and linear actuator, the same as the yoke pump.

The performance data for this pump was obtained from a computer program setup for a Univac in conjunction with other work in this area.

3. VANE PUMP

There are two types of variable delivery vane pumps. The first uses a concentric ring displaced from the center line of the drive shaft. The second uses a two-lobe ring. Pump delivery is varied by positioning the porting in a manner similar to the FAV pump.

(a) Concentric Ring

The concentric ring pump varies the displacement of the pump by varying the eccentricity of the ring and rotor. When the ring and rotor are concentric, there is no displacement. When the ring is offset from the rotor, volume between the vanes changes creating pump flow. The displacement of the ring is controlled by a linear actuator and a pressure sensitive control valve and return springs.

(b) Fixed Displacement Variable (FDV)

The FDV vane pump uses the conventional two-lobe balanced rotor type cam ring. (Figure 4) The delivery of the pump is varied in much the same manner as the FAV pump. The pump ring is rotated in relation to the porting allowing all or a portion of the fluid to be recirculated in the pumping cartridge.

The FDV vane pump is more suited to high pressure and high speed because the hydraulic forces on the rotor and drive shaft are balanced. The bearing and drive shaft of the two-lobe pump are much smaller than the single lobe pump. The size of the two-lobe pump is less as a result of the two-lobe design.

The performance data shown in Table I was obtained from a two-lobe fixed displacement vane pump.

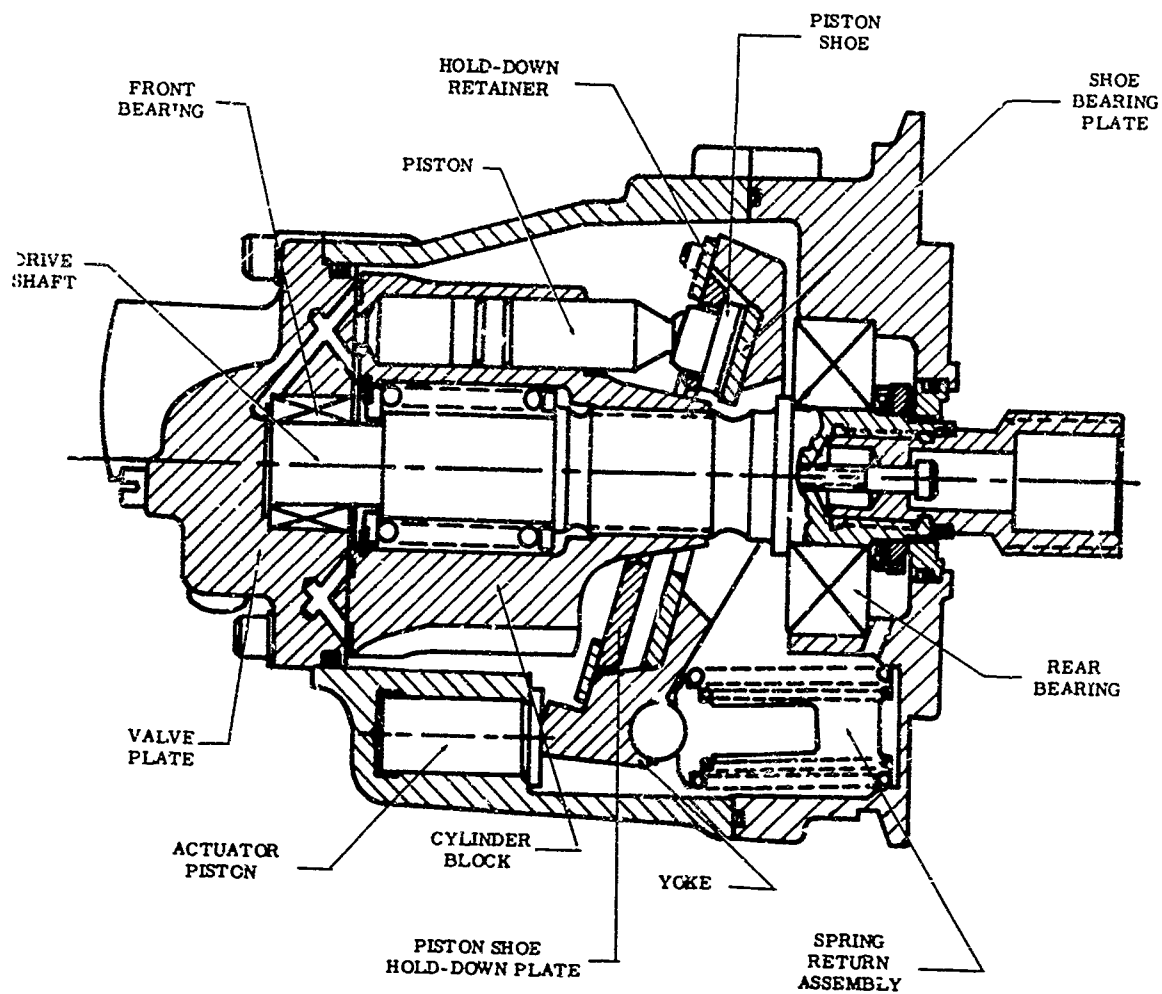


FIGURE 3 - INLINE PUMP

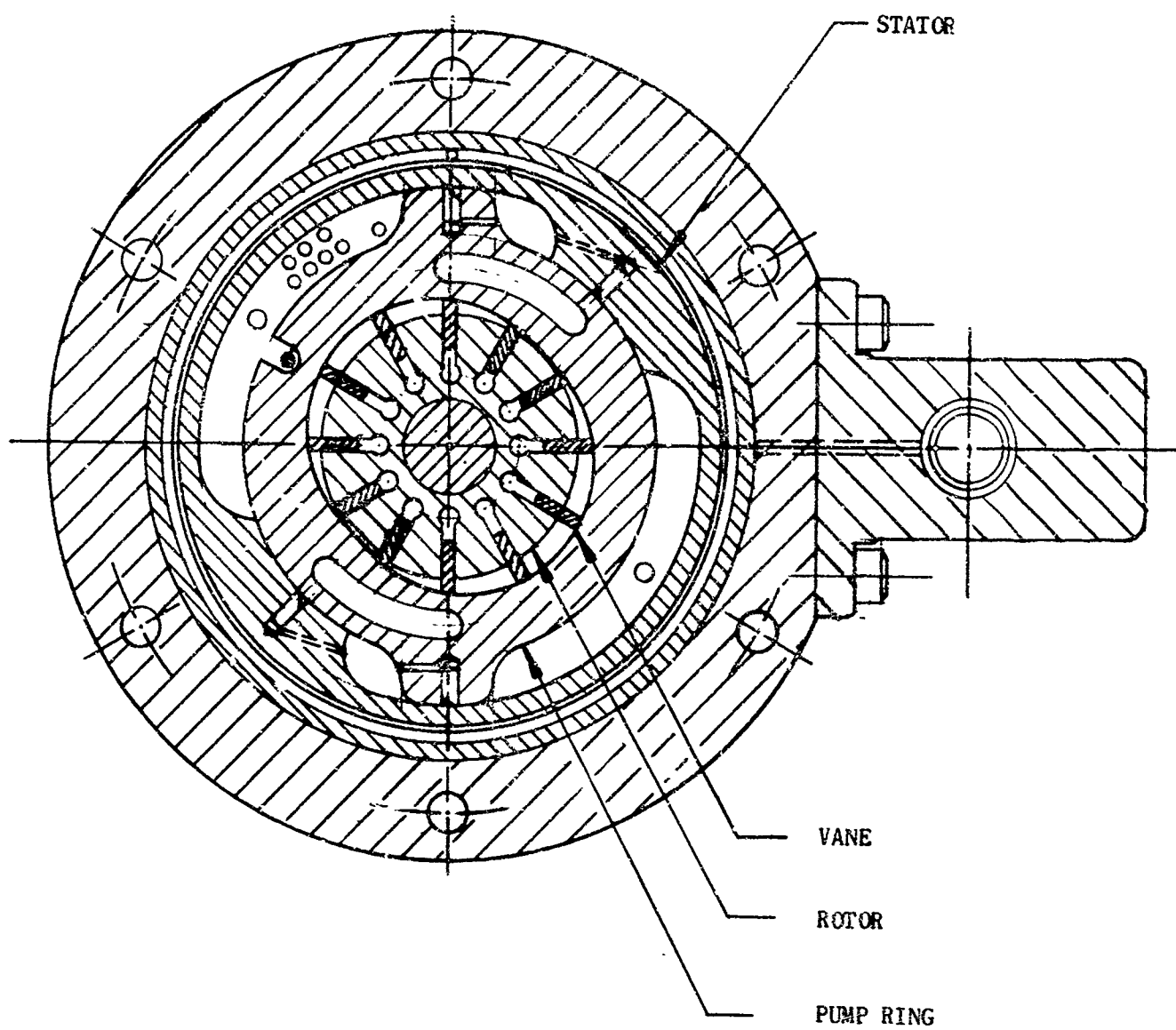


FIGURE 4 - VANE PUMP

4. ORBITAL PISTON PUMP

The orbital piston pump is different than most piston pumps in that the pistons slide in a rectangular annulus groove. (Figure 5) In the proposed design, there are two annulus track plates separated by a single drive plate. The annulus groove is circular and the pistons are about twice as long as they are wide. The annulus plates, drive plate, pistons and associated parts are assembled in a single rotating group package. The rotating group is mounted in the housing with one end on a pivot point and the other retained between the linear actuator and return springs.

When the center line of the shaft is on the same center as the annulus plate groove, the pistons rotate in the groove without relative motion between them. As the annulus plate is displaced to one side, the pistons become closer together on one side than the other. This difference in spacing creates a displacement and permits the pump to deliver fluid to the system.

The performance of the unit was extrapolated from actual test data obtained from a similar unit operating at lower rotational speed but at the same piston velocity.

5. RADIAL PISTON PUMP

Test data was available on three sizes of radial piston pumps, one of which is shown in Figure 6. The pump has nine pistons located in one plane. The maximum speeds at which the data was taken were one-half of the required speed equivalent to 18,000 rpm. The radial pump performance was lower than most of the other pumps analyzed. Therefore, it was not considered necessary to attempt to refine the results.

SUMMARY

After the power loss chart of Table I was constructed, it was reviewed to see which pump types could be eliminated from further consideration. As a result of this review, the following types were still in contention:

1. Bent Axis (Yoke)
2. Bent Axis (FAV)
3. In-Line
4. Orbital

A table of additional parameters was then constructed so that the pumps could be compared on a numerical basis. A brief explanation of the parameters shown in Table II follows:

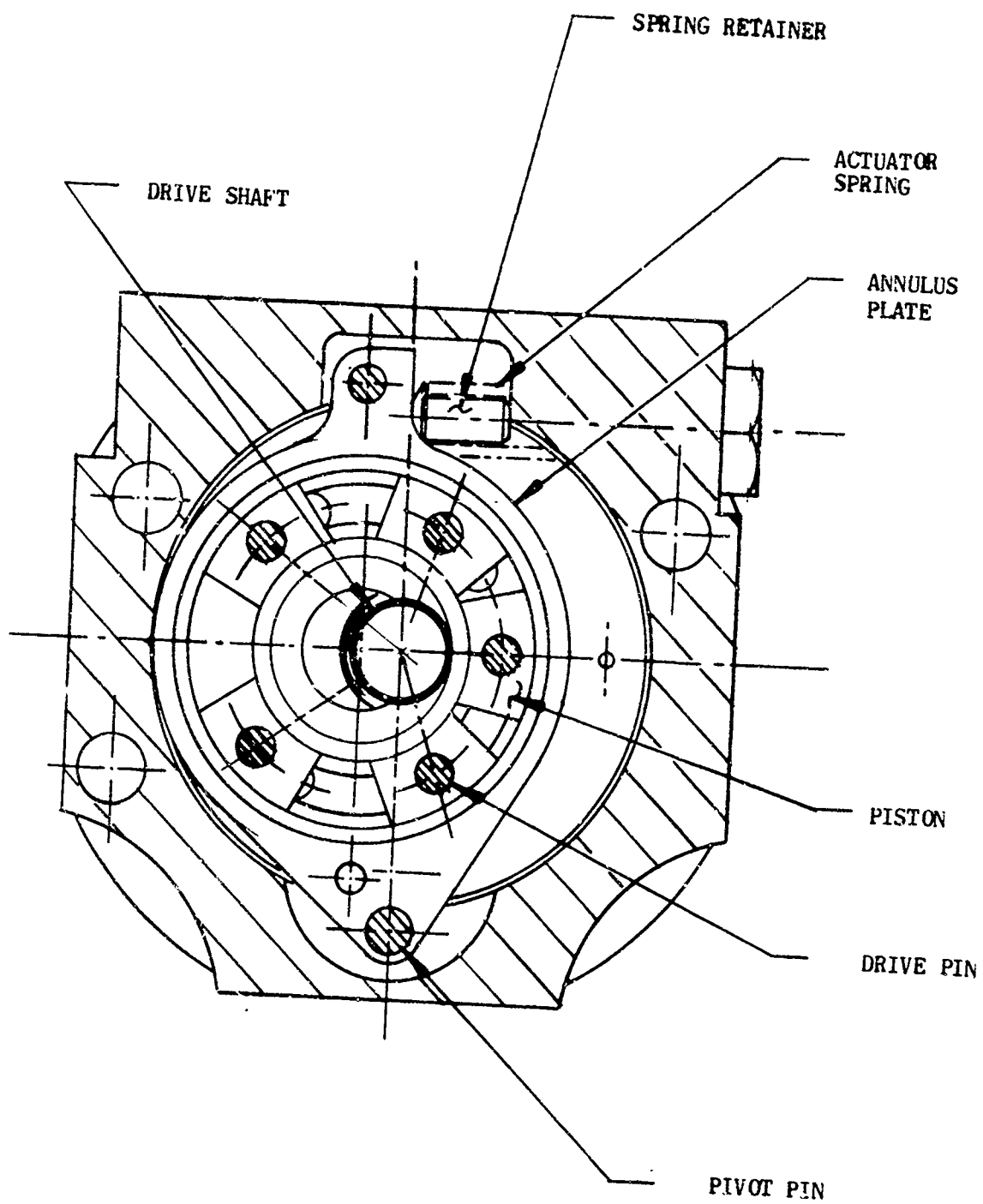


FIGURE 5 - ORBITAL PUMP

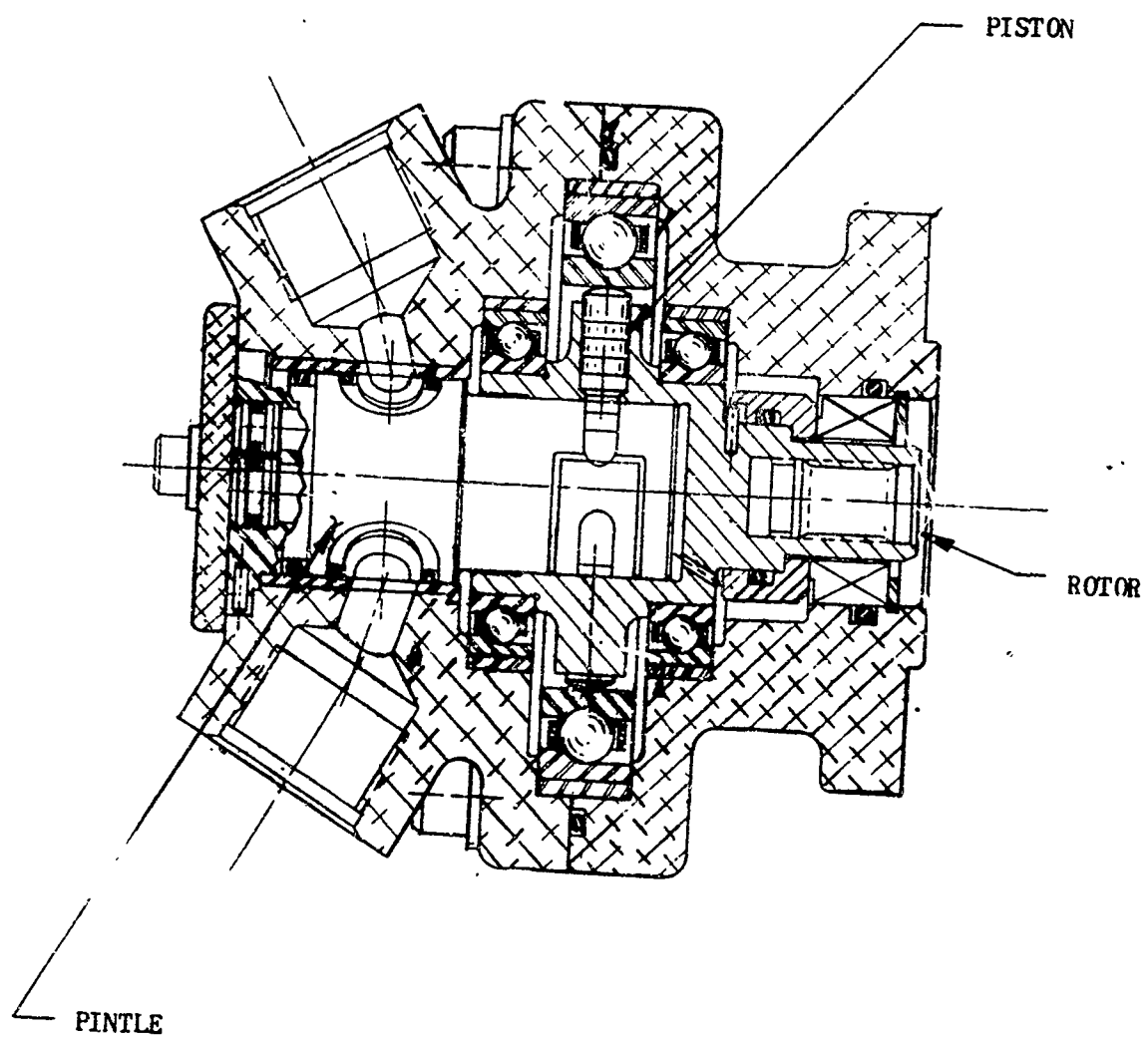


FIGURE 6 - RADIAL PISTON PUMP

TABLE II

COMPARISON OF PUMP PARAMETERS

<u>Design Parameter</u>	<u>FAV</u>	<u>Yoke</u>	<u>In-Line</u>	<u>Orbital</u>
1. Weight	9	3	9	9
2. Number of Parts and Cost	6	3	9	9
3. Envelope	6	3	9	9
4. Wear (Durability)	3	9	6	6
5. Performance	3	9	6	6
6. Pulsations	1	3	3	3
7. Control Stability	3	3	3	3
8. Stroke to Bore Ratio $\left(\frac{S}{D}\right)$	3	3	1	3
9. Displacement to Stroke Ratio $\left(\frac{Q}{S}\right)$	2	2	1	2
<hr/>				
Total	36	38	47	50
General Rating	4	3	2	1

1. Weight

For airborne equipment, low unit weight is desirable.

2. Number of Parts and Cost

The total number of parts bears a relationship to reliability. This is a general statement, and its complete validity depends on other factors. Cost was not a consideration, but there still remains some correlation between number of parts and total cost. The desired numbers should be low for both of these items.

3. Envelope

This implies both small envelope and the ability to fit within an envelope.

4. Wear (Durability)

This parameter refers to the probable difficulties in reaching the required life. Thus, a low rating implies either that the life requirements will be hard to reach or there are unknowns which keep the number low.

5. Performance

This is a numerical evaluation of Table I.

6. Pulsations

Pulsation level is a function of the pump and the system. The pulsation level will vary from full flow to zero flow. Taking a broad view, the FAV is the only one of the four pumps whose pistons continue to have a relative reciprocating motion when approaching zero flow. Thus, the ratings in the Table II reflect this reasoning.

7. Control Stability

Control stability, as in the case of pulsations, is a function of pump dynamics as well as the system. The oil selection (bulk modulus), the system damping and the system leakage are among the parameters to be considered. Control stability has been achieved in all of the pumps, the parameters to be adjusted are known, and the difficulties in reaching a satisfactory solution is judged to be about equal. Therefore, all of the pumps were given an equal rating.

8. Stroke to Bore Ratio

(See Item 9)

9. Displacement to Stroke Ratio

In general, pumps which have a large stroke to bore ratio and a small displacement to stroke ratio will have the best performance.

Table II was constructed using the above design parameters. A rating system of 1, 2 and 3 was used with 1 being a relatively poor rating, 2 a medium rating, and 3 the highest rating. In addition, the parameters were weighted by multiplying the basic rating by 3 for those characteristics that were considered to be the most significant. The entire table is an attempt to turn judgment, experience, test results and design know-how into a numerical evaluation from which the best pump choice may emerge.

It is to be noted in Table II that the in-line and the orbital have similar scores. Because of this, layouts and calculations were made for both pumps. The calculations revealed that the bearing area of the shoes on the pistons of the in-line pump would have to be increased over that of a lower speed unit. Accompanying the bearing area increase would be a general increase in the piston bore circle diameter and an increase in the diameter of the swash-plate which the shoes contact. The overall effect results in an increase in power losses.

As a result of the evaluation shown in Table II, plus the above information, the orbital pump was selected for further investigation.

SECTION III

DESIGN, DESCRIPTION AND OPERATION OF ORBITAL PUMP

The orbital pump design resulting from the study phase is described below:

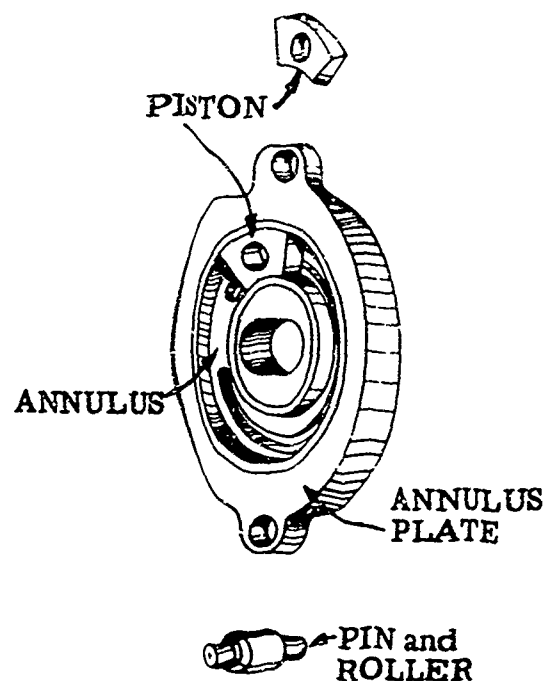
DESCRIPTION OF UNIT

The high speed engine pump has the following characteristics:

Theoretical Displacement:	.140 cu. in/rev.
Number of Pistons:	Ten (10)
Direction of Rotation:	Counterclockwise
Rated Outlet Pressure:	3000 PSI
Rated Shaft Speed:	15,000 RPM
Design Life:	500 Hours
Special Feature:	Variable Delivery
Overall Dimensions:	4¼" x 4½"
Mounting Flange:	AND 10261
Weight:	6.2 Pounds Dry

DESCRIPTION OF OPERATION

The Orbital Piston Pump derives its name from a set of curved pistons with a rectangular cross-section. The pistons "orbit" in a circular groove, or annulus. The annulus is in a non-rotating plate. The pump contains two annulus plates and two sets of pistons, one on each side of a rotating drive plate.



Holes in the pistons accept pins which are carried in slots in the drive plate. The drive plate, located between the two annulus plates, is splined and driven by the drive shaft.

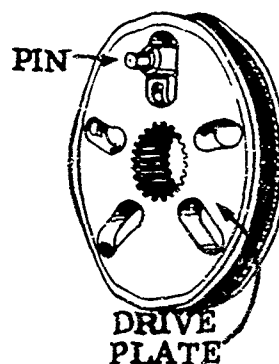


FIGURE 7 - ANNULUS AND DRIVE PLATES

Pumping motion of the pistons is achieved by eccentrically locating the center of the piston annulus groove with respect to the center of the drive plate. This geometric relationship produces a variation in the space between the pistons as they orbit through one revolution. The pumping action is achieved in this way:

As the pistons rotate from a position nearest to the drive plate center to a position farthest from the drive plate center, the spacing between the pistons increases. Rotation of the drive plate progressively decreases this spacing to its minimum again as the pistons move back to a position nearest the drive plate center and one cycle is completed.

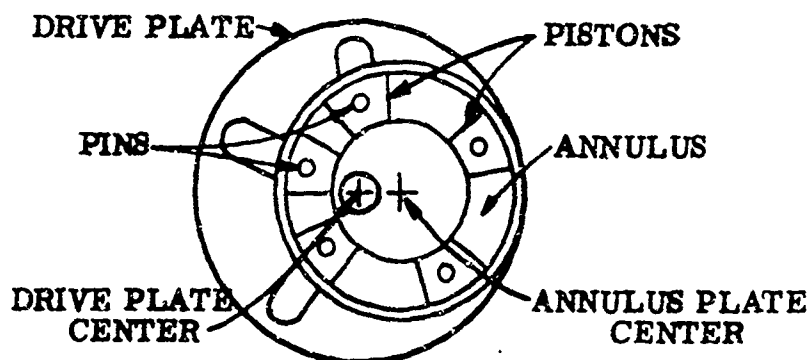


FIGURE 8 - DRIVE PLATE SUB-ASSEMBLY

The pump inlet or suction action is obtained by communicating or "valving" the section of the annulus associated with increasing piston spacing with a suitable inlet port. That portion of the annulus associated with decreasing piston spacing is communicated with a suitable discharge port, establishing the suction and pumping action.

ANNULUS PLATES

The annulus plates incorporate teflon seals in the outlet port. Discharge pressure acts to force the teflon seal lip against the face of the valve plate and acts to force the annulus plate against the drive plate.

This design assures positive sealing of high pressure fluid in the discharge port of the annulus plates. The thrust forces in the annulus plates are balanced against the drive plate, eliminating the need for costly, high-power-loss, thrust absorbing devices.

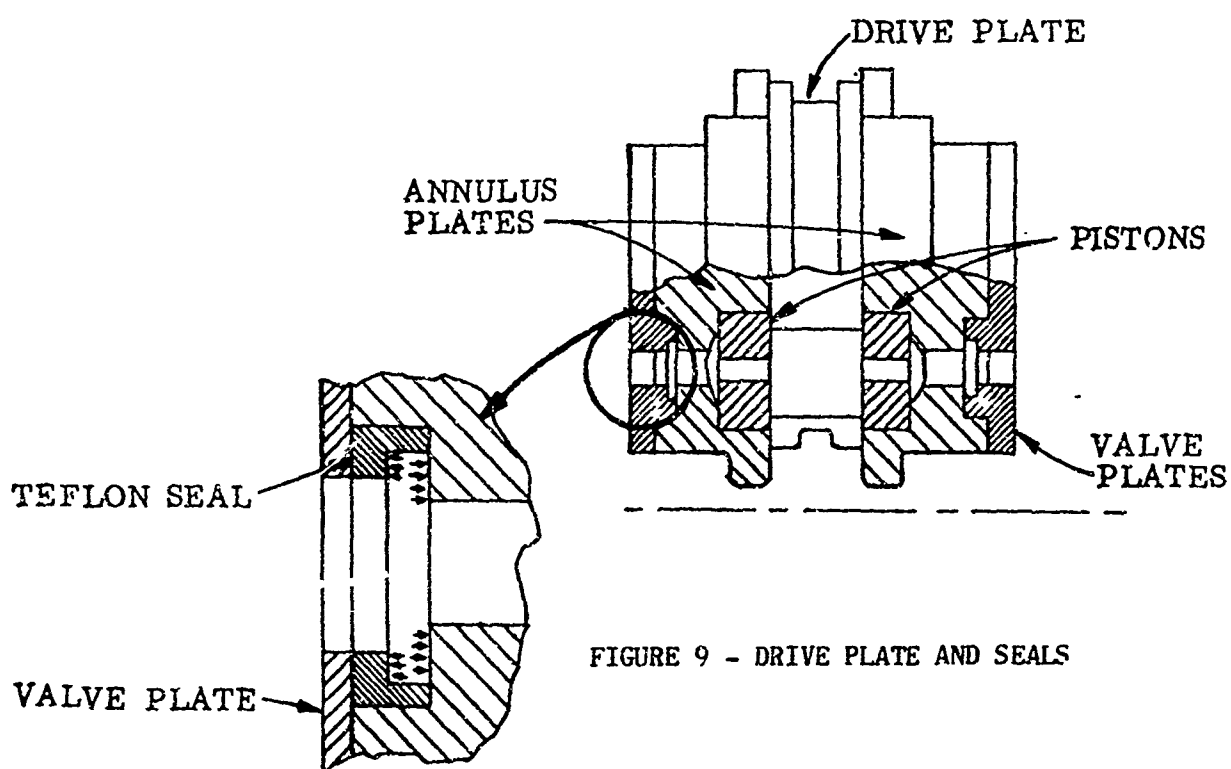


FIGURE 9 - DRIVE PLATE AND SEALS

Automatic pressure regulation is accomplished by incorporating a compensator valve and stroke actuator.

The valve senses variations in outlet pressure and reacts to provide hydraulic fluid to the stroke actuator (thereby reducing displacement) or to allow discharge of fluid from the stroke actuator (thereby increasing displacement).

The compensator valve spool is centered when the pump outlet pressure is at the pressure setting equivalent to the force exerted by the compensator spring. When the system flow demand increases, the outlet pressure reduces allowing the spring force to shift the spool to the left. This connects control pressure to case pressure, thus permitting the actuator spring to shift the annulus plates to a position of maximum pump displacement.

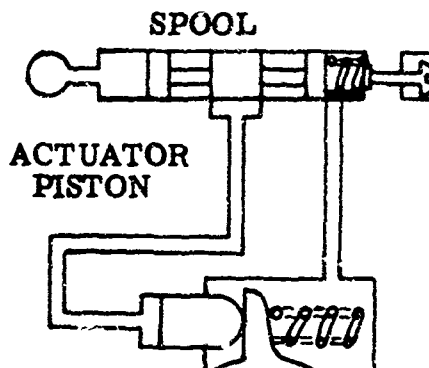
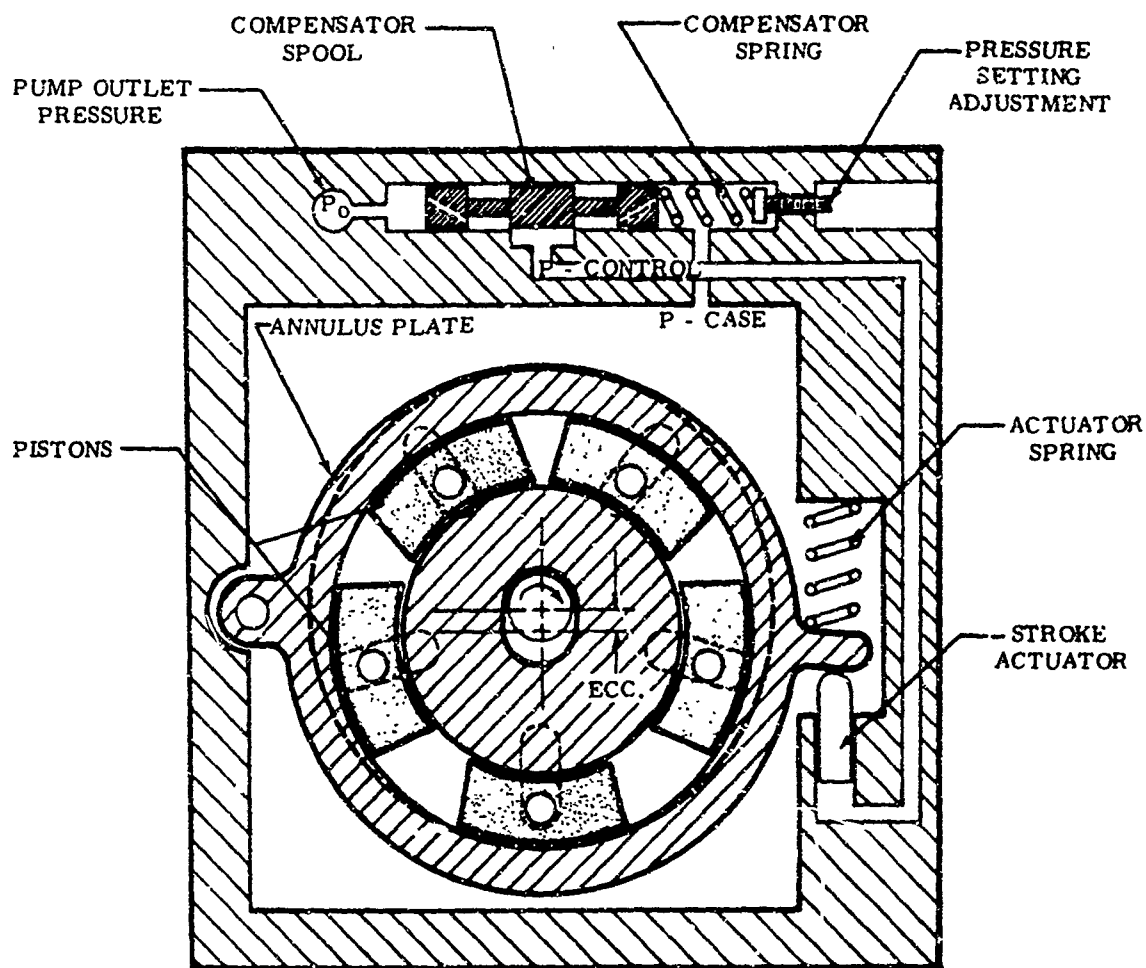


FIGURE 10 - ACTUATOR PISTON

As the load resistance increases, due to a reduction in system flow requirement, the outlet pressure exceeds the equivalent compensator spring force, the valve spool shifts to the right of the center position, allowing outlet pressure to act on the stroke actuator. When sufficient pressure is exerted on the actuator to overcome the force of the actuator spring, the annulus plates are shifted toward the zero displacement position (centers of drive plate and annulus plates approach coincidence). When the displacement is reduced sufficiently to equalize the forces on the valve spool (outlet pressure and compensator spring), the valve spool is recentered and hydraulically locks the stroke actuator to prevent further displacement change.

Variable output flow is provided by designing the annulus plates so that their centers can be varied with respect to the drive plate center and by controlling the magnitude of this eccentricity. This general arrangement of the pump is shown in a cross-section view below. The view is representative of a cross-section of the pump looking from the drive shaft end.



ANNULUS PLATE MOVES UP FOR DECREASING DISPLACEMENT
(FULL FLOW POSITION SHOWN)

FIGURE 11 - ANNULUS PLATE OPERATION

SECTION IV

FABRICATION PHASE

A test pump was fabricated and prepared for test. The assembled view of the pump is shown in Figure 12. The detail parts of the pump are shown in the exploded view, Figure 13. Figure 14 shows the assembly of the unit with the major parts and material combinations used.

Drawing 13861-K shows the piston and annulus plate sub-assembly. Note #2 on this drawing specifies the piston fit in the annulus plate. The initial fit of the pistons is line to line to .00015" clearance. The individual pistons and annulus plate are a matched set and not interchangeable.

Drawing 13878-K shows the rotating group sub-assembly.

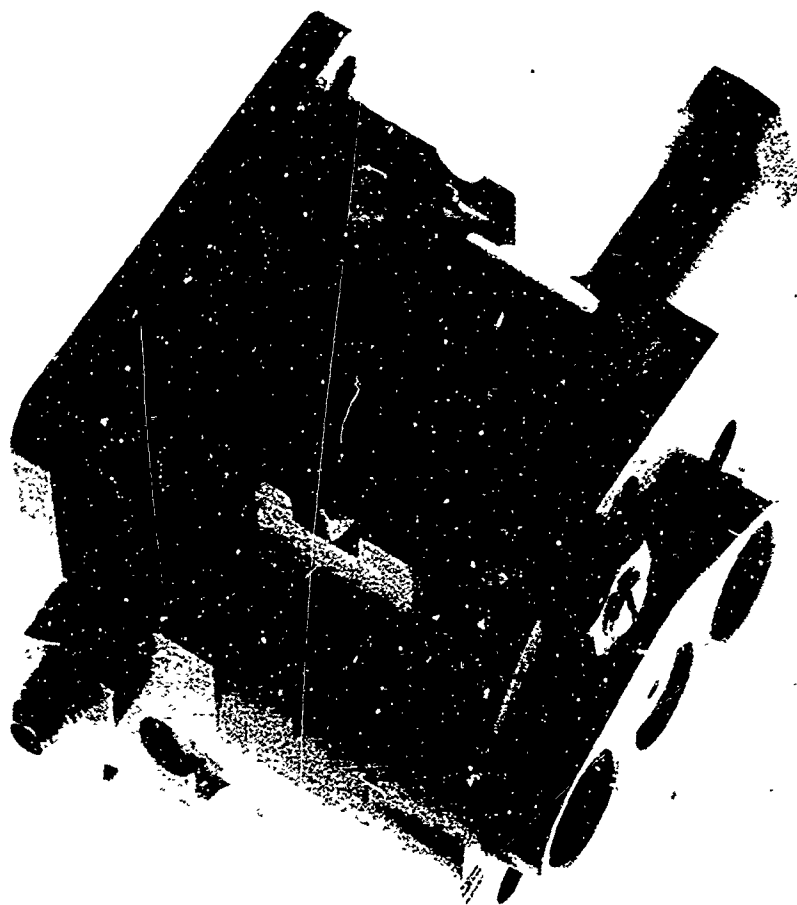


Figure 12 - Assembled Pump

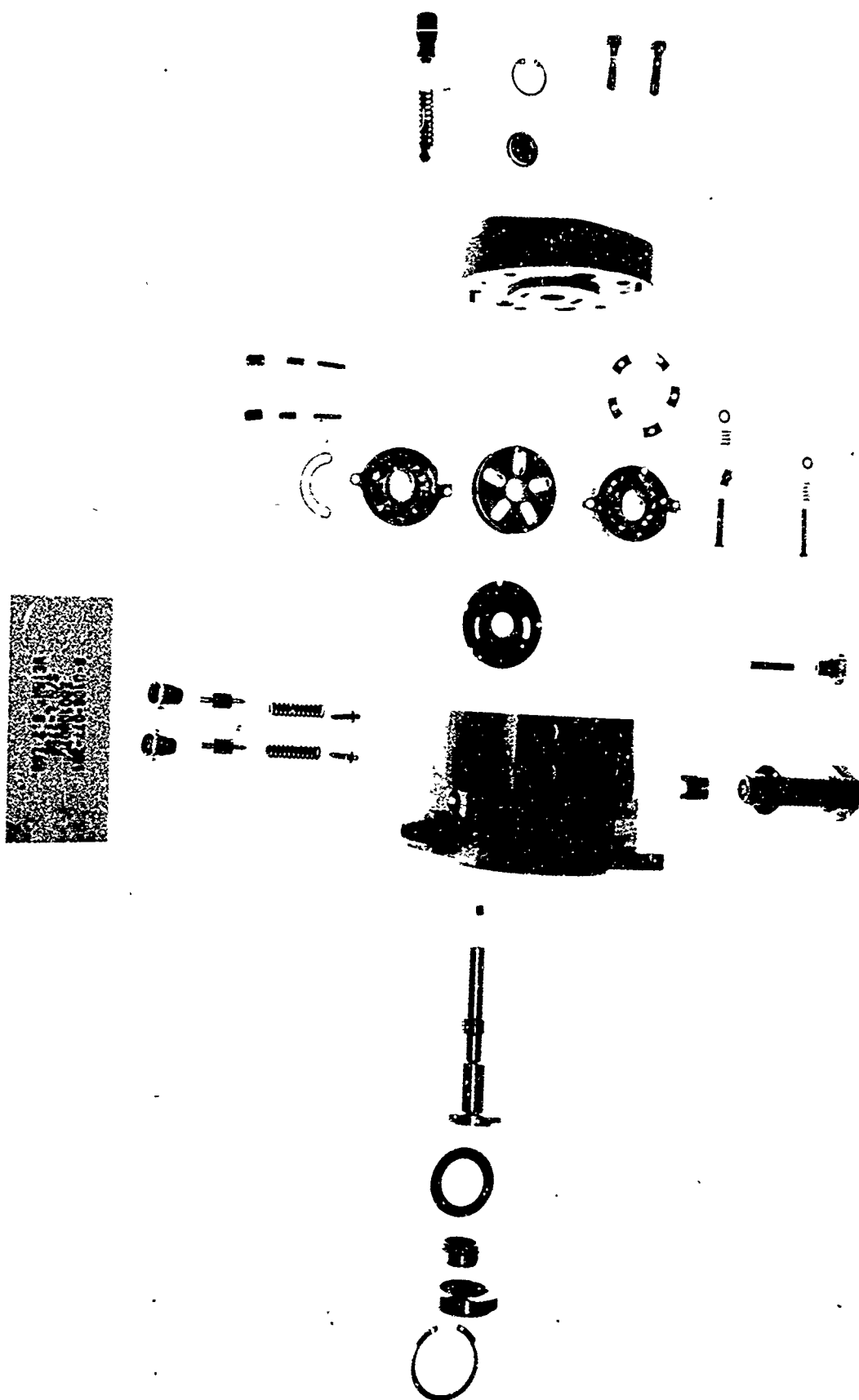


Figure 13 - Exploded View

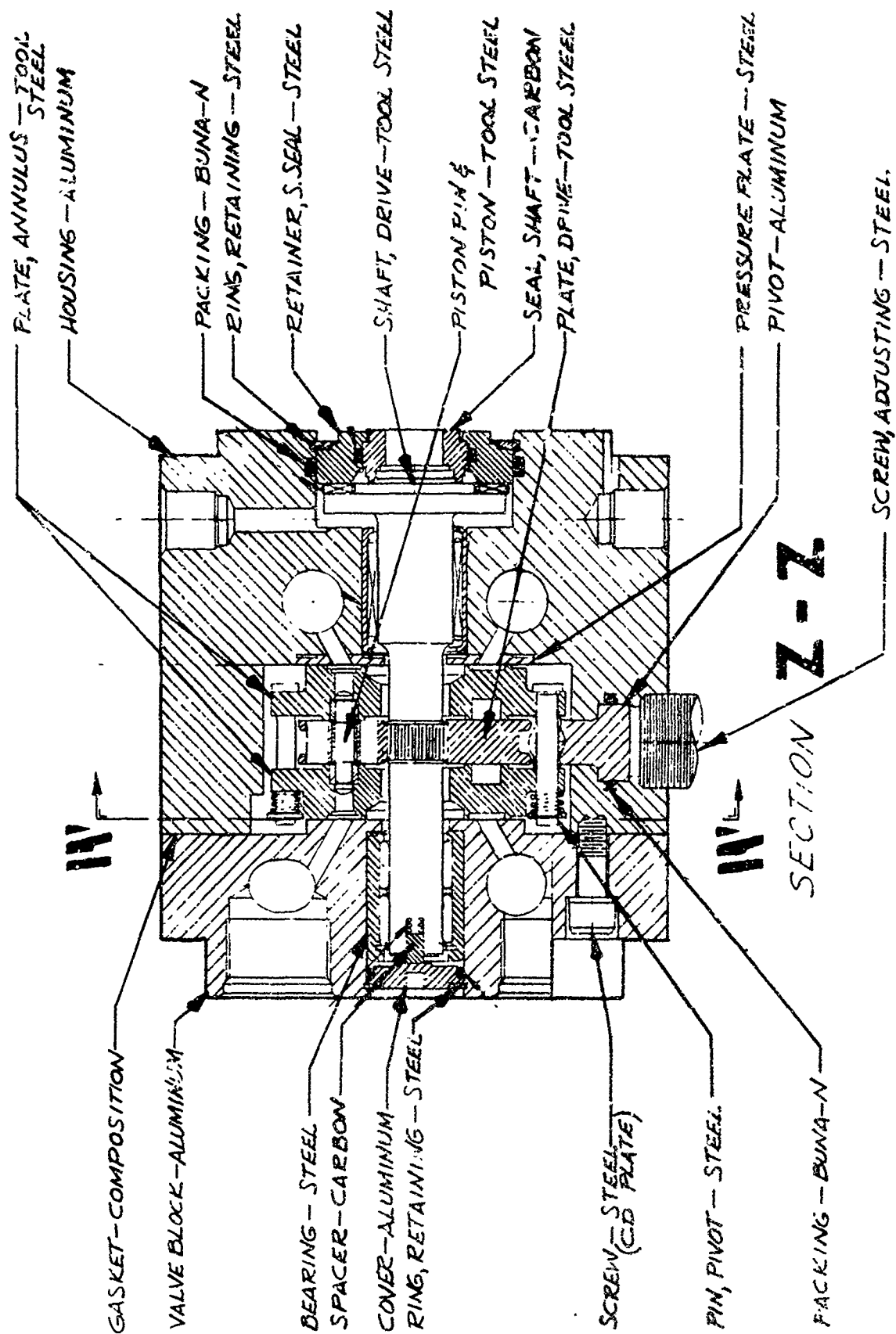
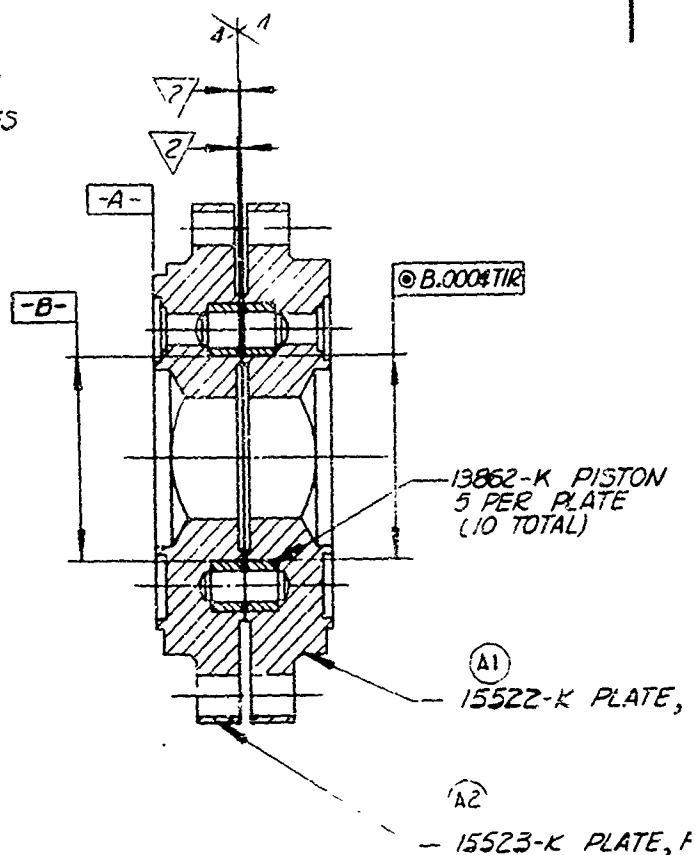
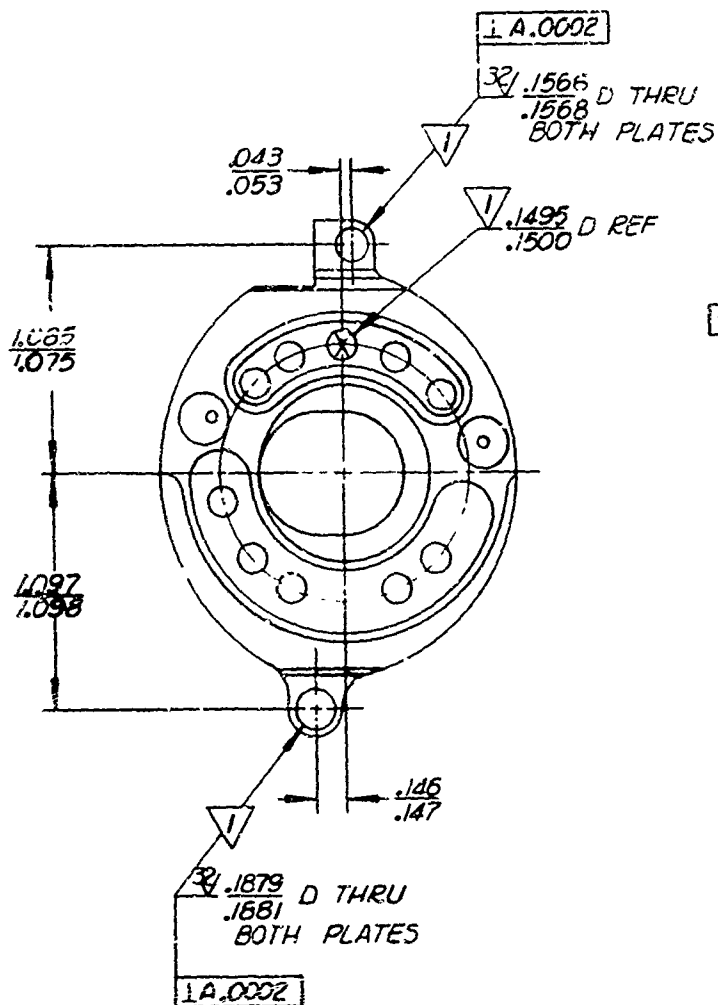


FIGURE 14 - ASSEMBLY WITH MAJOR PARTS IDENTIFIED

2

B



3. NOTE: PISTONS TO REMAIN WITH INDIVIDUAL PLATE AND MATCHED PLATES AND PISTONS NOT INTERCHANGEABLE

2 PISTONS TO FIT LINE TO LINE TO .00015 CLEARANCE WITH MATING PART

1 LOCATE PLATES FROM HOLE MARKED REF: FIXTURE PLATE

UNLESS OTHERWISE SPECIFIED	REF 1354C-K
SURFACE FINISH 125	DR PERIAN
2 PLACE DECIMALS $\pm .03$	CHK 1/2
3 PLACE DECIMALS $\pm .010$	T O REL 1/2
ANGULAR DIMENSIONS $\pm 2^\circ$	PROTO REL
CONCENTRICITY .010 TIR MAX.	PROD REL
REMOVE ALL BURRS & BREAK SHARP EDGES .010 MAX.	STDS REVIEW
MACHINE PER VS 1-3-3-1	MFG REVIEW
MATERIAL	ENGRG APP
HEAT TREATMENT	APP

NEXT ASSY USED ON APPLICATION

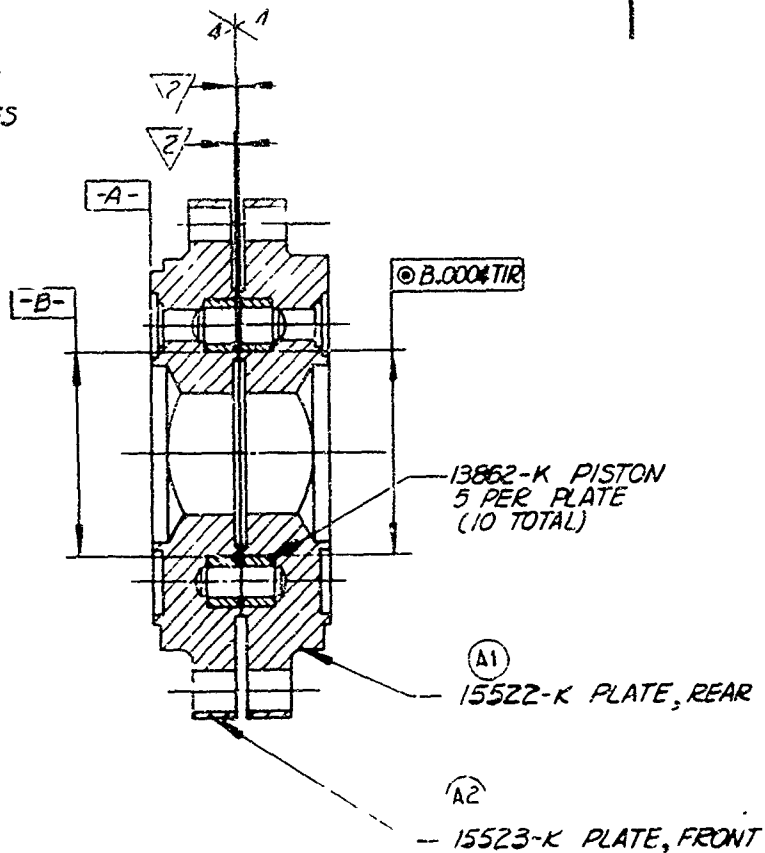
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REVISIONS				
SYM	ZONE	DESCRIPTION	DATE	APPROVED
A	1A	1. 15522-K WAS 13858-K	7-24-60	
	1A	2. 15523-K WAS 13867-K	T.M.	F. J. Sullivan

THRU
PLATES

REF



13861-K

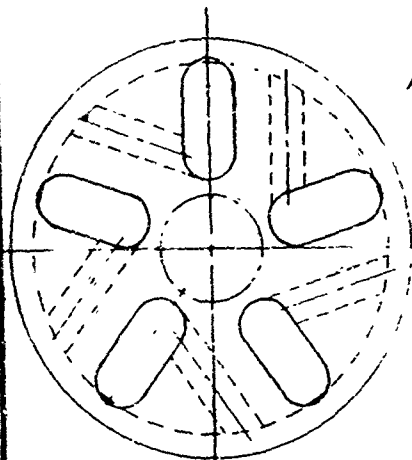
A

VS 1-3-3-1

APPLICABLE
SPECIFICATIONS

UNLESS OTHERWISE SPECIFIED		REF 1354C-K		VICKERS INCORPORATED	
SURFACE FINISH 125 ✓		DR PERIAN		DIVISION OF SERRY RAND CORPORATION	
2 PLACE DECIMALS ± .03		CHK 1/1/60		DETROIT, MICH. TORRANCE, CALIF.	
3 PLACE DECIMALS ± .010		DATE 1-29-65		AEROSPACE DIVISION	
ANGULAR DIMENSIONS ± 2°		T O REL 2957		TITLE	
CONCENTRICITY .010 TIR MAX.		DATE 1-29-65		PISTONS AND PLATES	
REMOVE ALL BURRS & BREAK SHARP EDGES .010 MAX.		PROTO REL		SUB-ASSEMBLY	
MACHINE PER VS 1-3-3-1		PROD. REL		(140)	
MATERIAL		STDS REVIEW		CODE	
HEAT TREATMENT		MFG REVIEW		IDENT NO.	
NEXT ASSY USED ON APPLICATION		ENGRG APP		62983	
		APP		DWG SIZE	
				C	
				13861-K	
				SCALE	
				WT ACT. CALC SHEET	

B



SECTION Z-Z

SHOWING DRIVE PLATE ONLY

131697 RETAINING RING

13868-K SLEEVE
5 REQD

120722 RETAINING RING

14150-K RETAINER

98652-X SPRING

15812 -K DRIVE PLATE

14153-K ACTUATOR PIN

15813-K ROLL
5 REQD

13867-K PISTON
5 REQD

13861-K PISTONS & PL

14152-K PIVOT PIN

14195-K PIVOT

98651-X SPRING

14151-K RETAINER

(B)

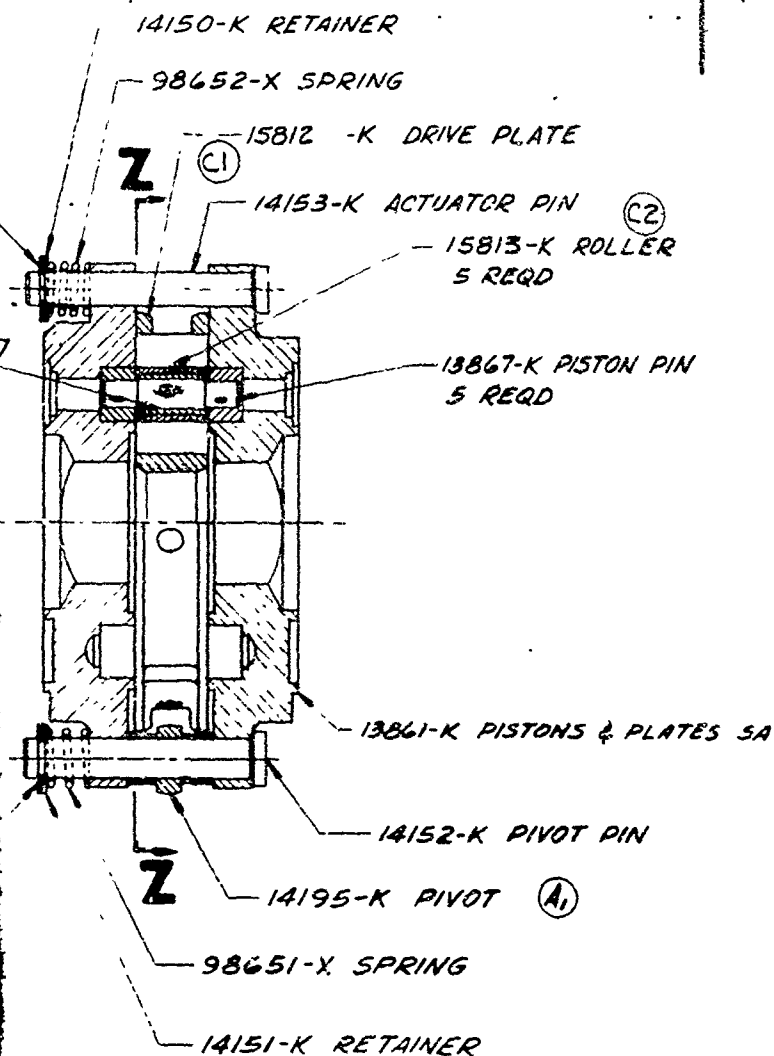
FIT SLEEVES TO PROVIDE .0002 .0003 PISTON
TO DRIVE PLATE CLEARANCE.
SLEEVES SIDES TO BE PARALLEL WITHIN
.00005

13575-K

NEXT ASSY USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED	REF
SURFACE FINISH 125/	DR SU
2 PLACE DECIMALS ± .03	CHK EN
3 PLACE DECIMALS ± .010	TO REL
ANGULAR DIMENSIONS ± 2°	PROTO REL
CONCENTRICITY .010 TIR MAX.	PROD REL
REMOVE ALL BURRS & BREAK SHARP EDGES .010 MAX.	STDS REVII
MACHINE PER VS 1-2-3-1	MFG REVIE
MATERIAL	ENGRG APP
HEAT TREATMENT	APP

REVISIONS				
SYM	ZONE	DESCRIPTION	DATE	APPROVED
A		EC - 983-986 1 WAS 13864-K PIVOT 2 ADDED SECT Z-Z	2-19-65	J. W. Parnell
B		1. NOTE V ADDED	3-20-65 T. J.	Fullerton
C	2B 1B	(1) 15812-K WAS 15852-K (2) 15813-K WAS 13869-K	3-20-65 T. J.	Fullerton



13878-K

VS 1-3-3-1

APPLICABLE SPECIFICATIONS

UNLESS OTHERWISE SPECIFIED		REF		VICKERS INCORPORATED	
SURFACE FINISH 125		DR SUN	DATE 1-28-65	DIVISION OF SPERRY RAND CORPORATION	
2 PLACE DECIMALS ± .03		CHK F. W. PERIAN	DATE 1-29-65	DETROIT, MICH. TORRANCE, CALIF.	
3 PLACE DECIMALS ± .010		T O REL PASS	DATE 1-29-65	TITLE	
ANGULAR DIMENSIONS ± 2°		PROTO REL	DATE	ROTATING GROUP	
CONCENTRICITY .010 TIR MAX.		PROD. REL	DATE	SUBASSEMBLY (.140)	
REMOVE ALL BURRS & BREAK SHARP EDGES .010 MAX.		STDS REVIEW	DATE	CODE IDENT NO.	
MACHINE PER VS 1-3-3-1		MFG REVIEW	DATE	62983 C	
MATERIAL		ENGRG APP	DATE	DWG SIZE	
HEAT TREATMENT		APP	DATE	13878-K	
NEXT ASSY USED ON APPLICATION		SCALE 2/1		WT ACT. - CALC - SHEET	

2

SECTION V

EVALUATION TEST OF ORBITAL PISTON PUMP

The tests conducted on the orbital pump consisted of the following: break-in, performance, sampling and low temperature tests.

The type of tests and procedures used were in accordance with MIL-P-25868. The test procedures and equipment used were as follows:

Break-In

The break-in was run on a 15 horsepower variable speed electric drive. The circuit used was the same circuit used for the performance test. A schematic of the circuit is shown in Figure 15.

Performance Test

The performance test was run on a 50 horsepower variable speed electric drive. A schematic of the circuit used is shown in Figure 15. The method of controlling pump discharge pressure was changed to a Vickers relief valve as shown in the schematic.

Sampling Test

The sampling test circuit is similar to the performance circuit except a cycling circuit was added. The cycling circuit consists of an electrically operated, timer controlled, directional valve and a relief valve. The relief valve is used to control discharge pressure. (See Figure 16)

Low Temperature Test

The low temperature start tests were conducted in a refrigerated test chamber that maintained an ambient temperature of $-65^{\circ}\text{F.} \pm 5^{\circ}\text{F.}$ A simplified circuit for these tests was used. It included a Vickers relief valve for a load valve and a .95 cu. in/rev. piston type meter motor. The circuit is shown in Figure 17. The drive motor for the test was a fixed displacement hydraulic motor driven by an auxiliary power supply. All pressure readings were recorded on a six-channel Sanborn recorder. Continuous readings were taken for all cold starts. Acceleration of the test unit was determined by the input flow to the hydraulic drive motor. The input to the motor was controlled by a manually operated valve.

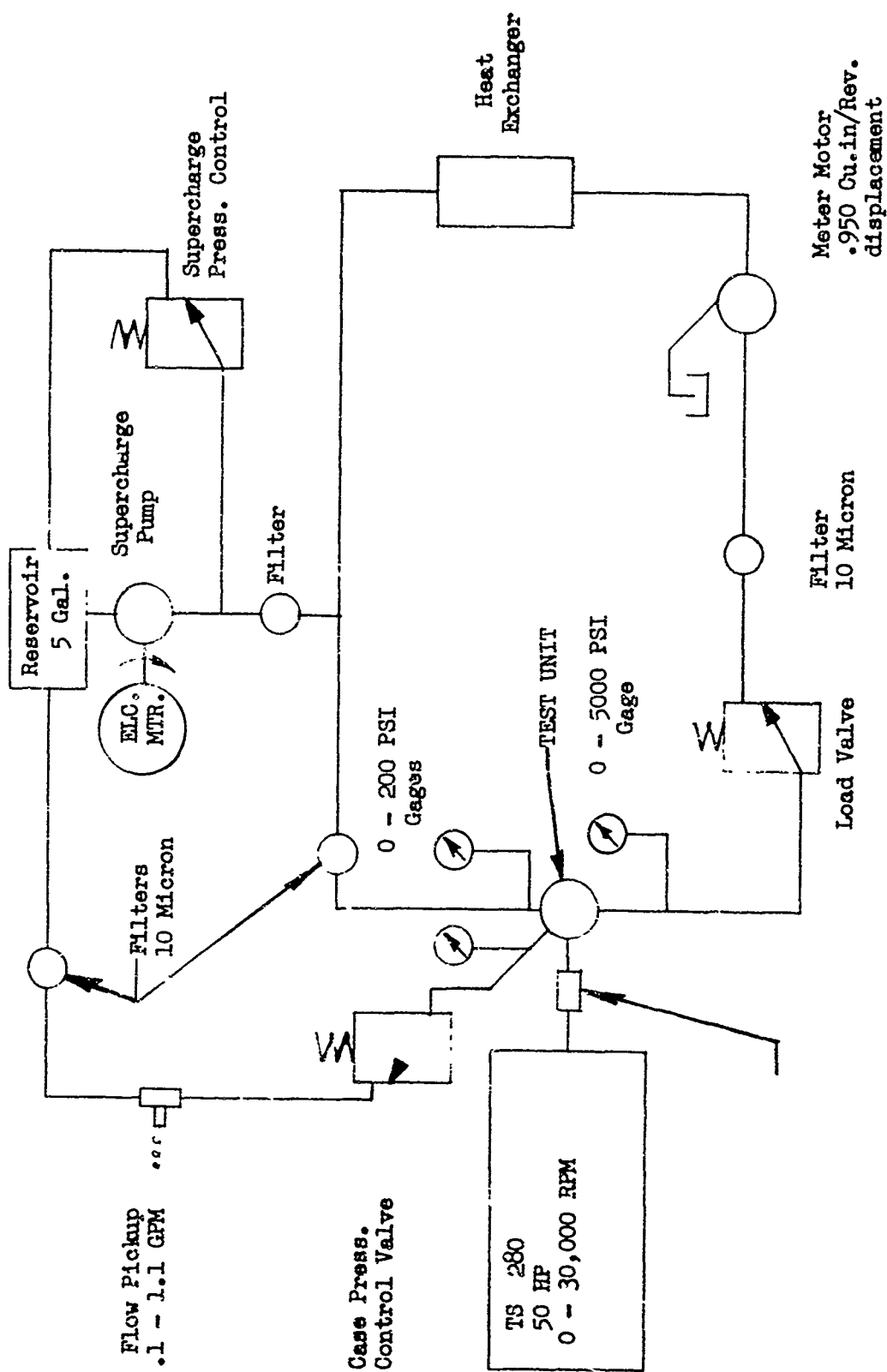


FIGURE 15 - PERFORMANCE TEST CIRCUIT DIAGRAM

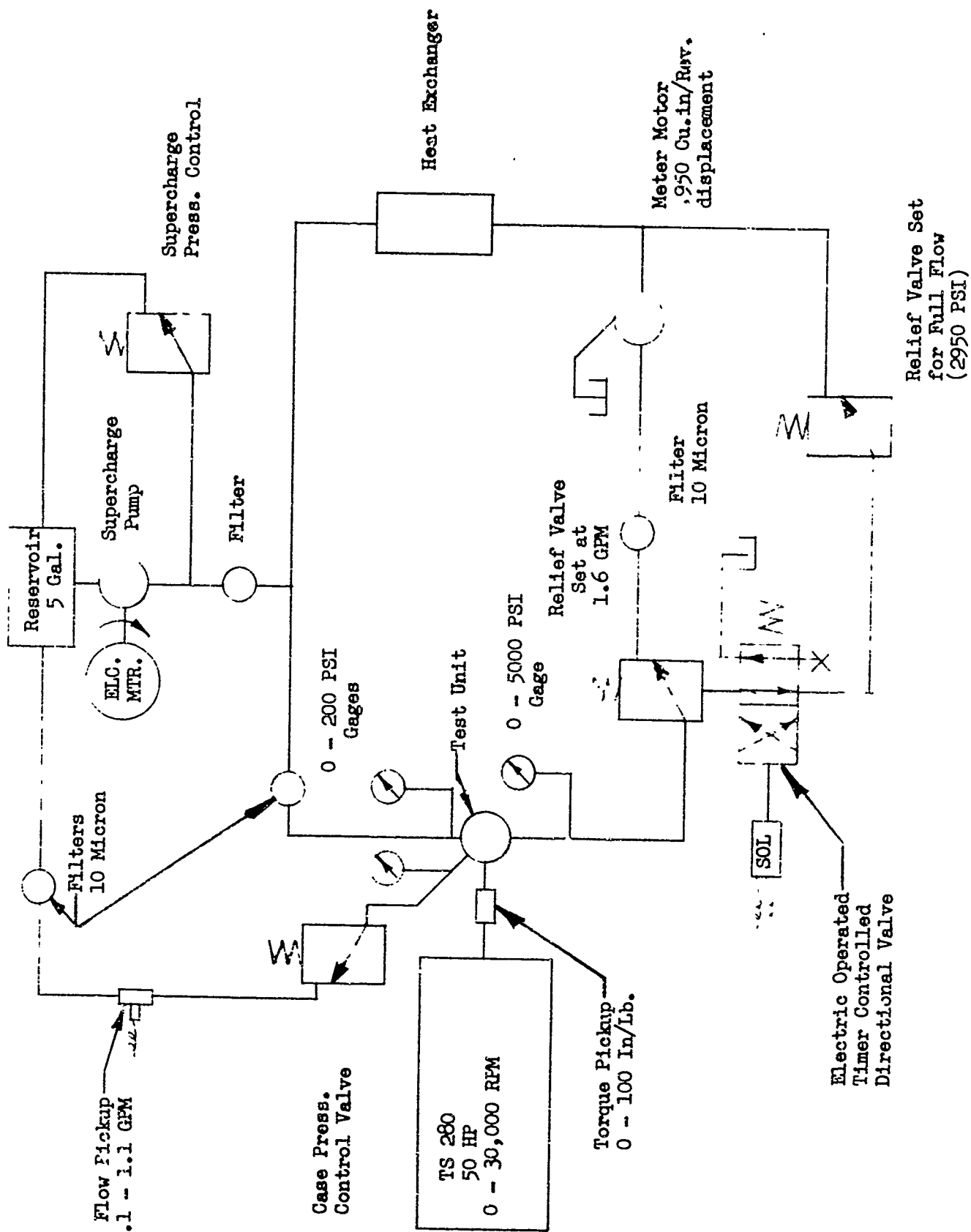


FIGURE 16 - SAMPLING TEST CIRCUIT DIAGRAM

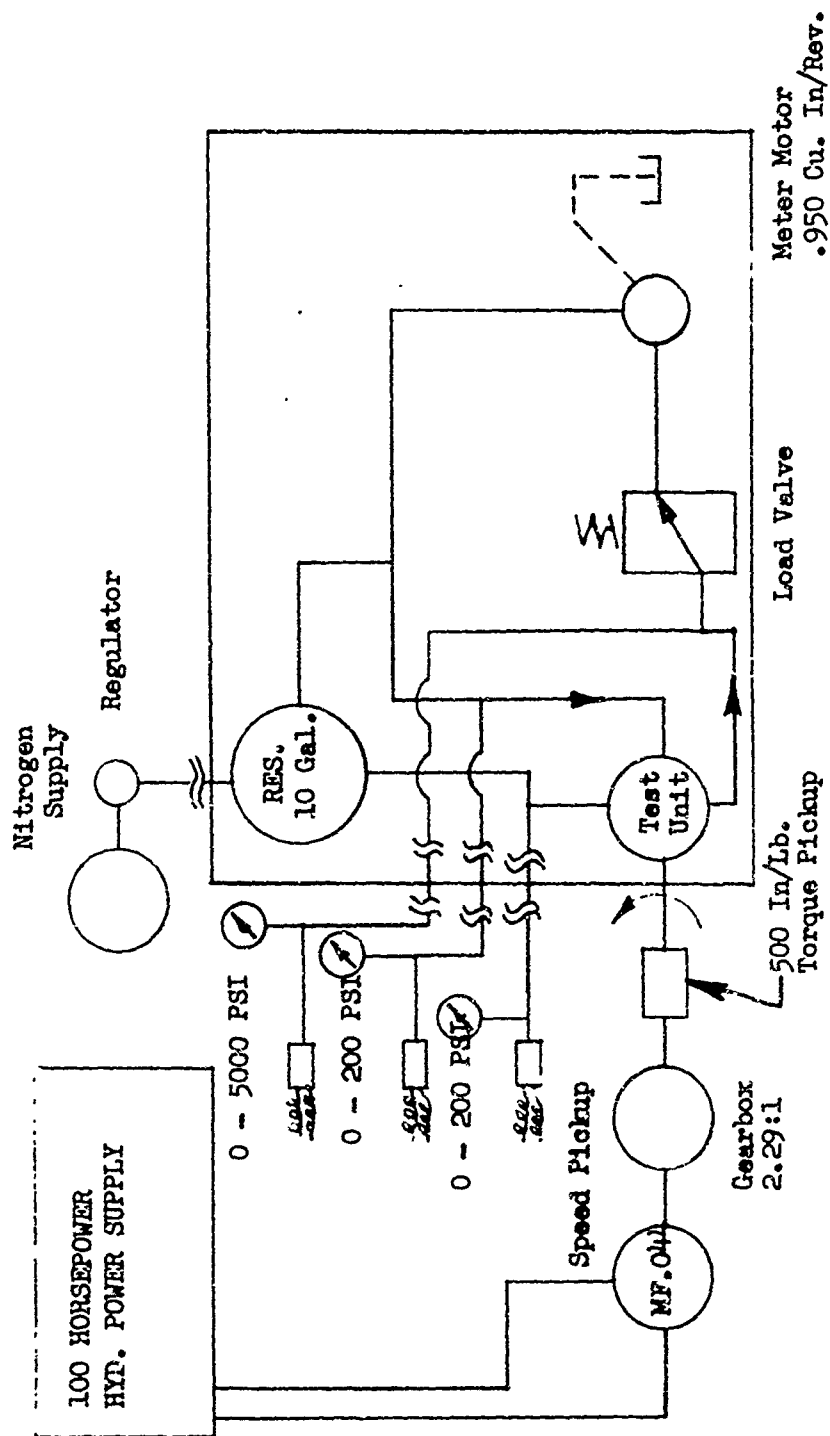


FIGURE 17 - COLD START, TEST CIRCUIT DIAGRAM

SUMMARY OF TESTING

Break-In

The break-in run was restricted to a maximum of 15,000 RPM, 2800 PSI and 8.6 GPM because of test stand limitations. During the break-in, the pump discharge pressure was controlled by a manual load valve. A preliminary check of pressure pulsations was made during the break-in run. From 1800 to 4000 RPM the pressure pulsations were between ± 50 PSI to ± 75 PSI. From 7400 RPM to 12,000 RPM, the pressure pulsations were between ± 50 PSI to ± 125 PSI.

Performance Test

The performance tests were run at full delivery (10.8 GPM), full pressure (2800 PSI), and at rated pump speed (18,000 RPM). The fluid inlet temperature was maintained at $137 \pm 5^\circ\text{F}$. The overall efficiency at these conditions was 68%. Tests with a second set of annulus plates were made. The unit was run at 18,000 RPM, 10.7 GPM, 3000 PSI and 137°F . The overall efficiency of the second test was 70.1%. The overall efficiency was lower than the design calculations and interpolation of other orbital pump test data indicated it should be.

Horsepower Loss Investigation

In an effort to determine the cause for the low overall efficiency, a series of power loss tests was conducted. The power loss tests included both mechanical and hydraulic power losses. See Figure 18. The first test consisted of running a pump with only the drive shaft bearings and shaft seal assembled. For the second test, the drive plate and annulus plates were added. The pistons and piston rollers were added for the third test. The final test was run as a completely assembled unit at both minimum flow and full flow at minimum pressure and maximum pressure.

The results of these tests showed that 45% of the power loss was caused by the rollers moving in the drive plate. 25.6% of the power loss was caused by friction between the drive plate and annulus plates. These two areas accounted for the majority of the power loss.

The rotation of the pistons and rollers accounted for 15.4%. The drive shaft bearings and shaft seal accounted for 7.6%. The remaining 6.4% was attributed to hydraulic

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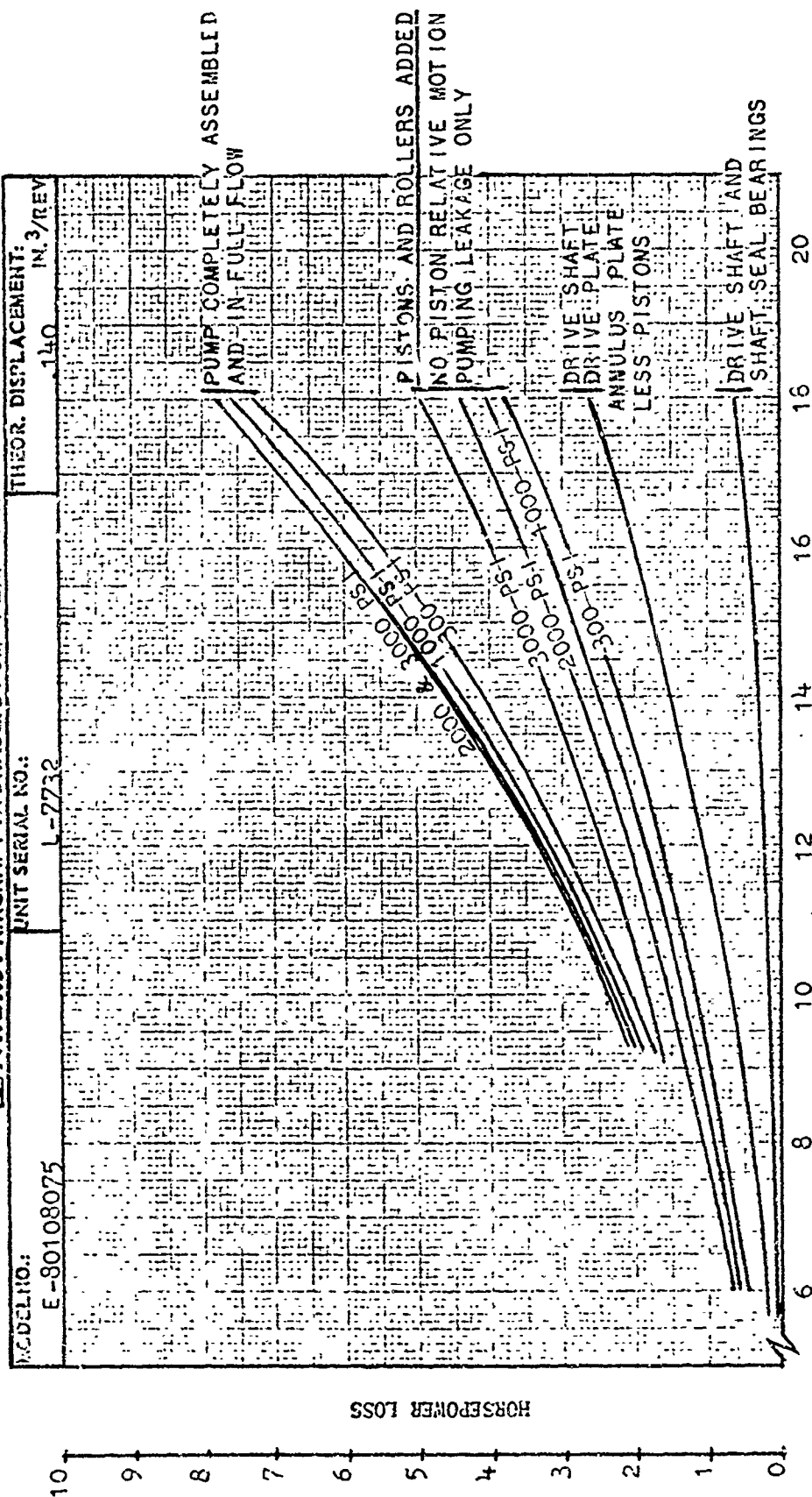


FIGURE 18 - HORSEPOWER LOSS CURVE

SPEED: AS NOTED TEMPERATURE: 95°F FLUID: MIL-H-5606 INLET PRESSURE: 60 OUTLET PRESSURE: CASE PRESSURE: BACKPRESSURE: CALCULATED DISPL.: .14285		REMARKS HORSEPOWER LOSS VERSUS SPEED		CURVE NO. 8-0108-078-310-03 DATE 8-17-66 APPROVAL 1. <i>HAH</i> 2. <i>660818</i> 3.
DATA SHEETS:		NOTE: These data reflect performance of one pump, and must not be considered representative of average or guaranteed values for production pumps.		

Horsepower Loss Investigation (Continued)

losses and other miscellaneous causes.

The highest percentage of horsepower loss (45%) was caused by the relative motion between the piston rollers and the drive plate. This motion is inherent in the design and is proportional to the pump stroke. At minimum flow, the movement between the rollers and drive plate is small and the horsepower losses are low. As the displacement is increased at a constant pressure, the losses increase proportionally to pump stroke.

The rotation of the drive plate against the annulus plates accounts for approximately 25.6% of the total horsepower loss. The drive plate and annulus plates contact only at the seal lands. This contact cannot be eliminated but the load, hydraulic and mechanical, can be minimized to reduce the friction. This was accomplished when a modification of the unit was made. However, further development work in this area is necessary.

An investigation of non-rotating parts and porting configuration was conducted to determine if these parts contributed to the horsepower loss. The tests indicated they had no significant effect. The pump housing to drive-plate clearance was increased to change the viscous friction influence. This did not decrease the horsepower loss.

Functional Test

The unit completed the functional test without change in flow or torque. The inlet temperature varied from 130°F. to 220°F. The overall efficiency at 15,000 RPM, 2950 PSI was 71.2%.

Calibration Test

The calibration of the unit was completed for 25, 50, 75, and 100% of rated speed. Readings were taken at 100, 75, 50, 25 and 0% of full flow. On the last reading of the calibration test, at 110% of rated speed (19,800 RPM), the unit failed.

The cause of the failure was piston binding in the annulus groove. During operation at high speed, small particles of silver were removed from the annulus grooves.

Calibration Test (Continued)

Some of these particles became partially trapped in cavitation pits causing a tight spot on the annulus groove.

The impact load at this point on the piston rollers caused the rollers to crack. Part of the broken roller in one slot wedged between the pin and the drive plate slot end breaking the drive plate. See Figure 19.

The corrective action taken was to increase the piston clearance in the annulus groove to .0002". The sleeve clearance was increased from .0001/.0002" to .0002/.0003". At this time, the pump had 57 hours of operation; 16 of these hours were at 18,000 RPM. The total time on the pump including the horsepower loss investigation was 57 hours.

Rebuild and Break-In

The pump was rebuilt and the test resumed. The unit failed during break-in while operating at 3000 RPM and 1200 PSI. The cause of failure was piston seizure in the annulus track.

Rebuild and Break-In

The unit was rebuilt with piston clearance increased to .0008" to eliminate the possibility of piston seizure. After completion of break-in and calibration, the piston clearances had increased beyond the allowable limits due to creeping of the silver plate in the annulus groove.

An investigation was started to determine the proper piston clearance to eliminate piston seizure. This was correlated with acceptable unit volumetric efficiency. The creeping of the thick silver plating (.003" to .005") could not be prevented by increased piston clearance. To eliminate the plating difficulty, the following three changes were made:

- (1) The silver plating of the annulus groove was reduced to .0002"/.0004" thickness.
- (2) The timing and metering of the ports was changed.
- (3) The hydraulic balance of the annulus plates was changed to reduce wear and friction.

The unit was then rebuilt with the changes as recommended.

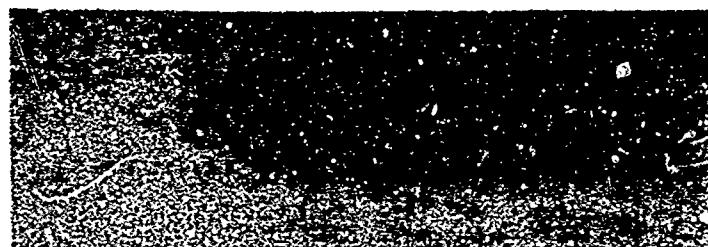
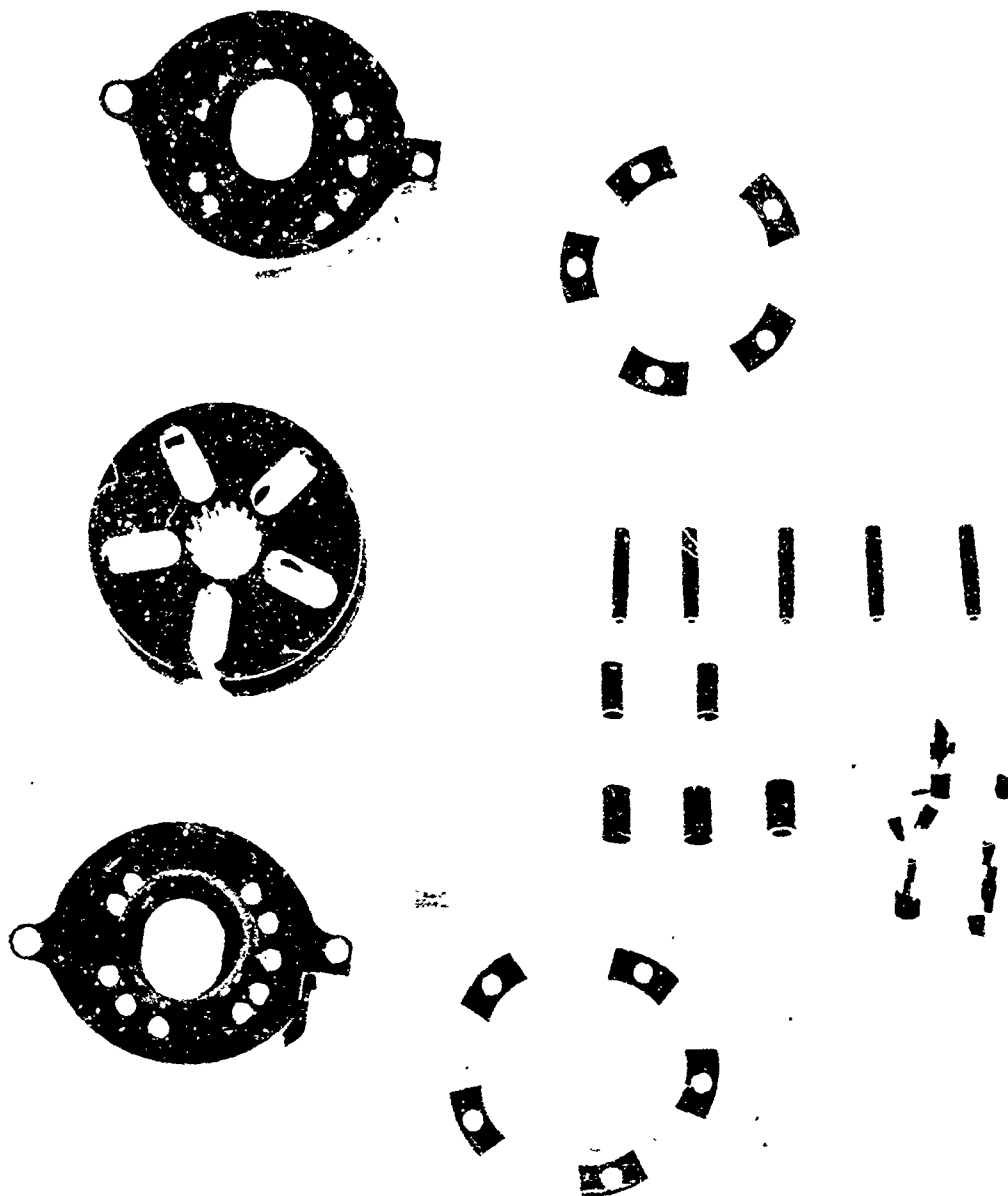


Figure 19 - Failed Rotating Group

Rebuild and Break-In (Continued)

A review of the design and the original requirements resulted in a change of specifications. The 1000 hour endurance life was changed to 500 hours. The maximum speed of 18,000 RPM was reduced to 15,000 RPM and the flow was reduced from 10.0 GPM to 8.3 GPM.

The unit completed break-in and calibration test without incident. The unit was calibrated at 25, 50, 75, 100 and 110% of rated speed (15,000 RPM). The overall efficiency at 15,000 RPM, 2950 PSI and 124°F. inlet temperature was 71.2%.

Proof Pressure

The overall efficiency of the unit before proof pressure test was 71.6% (at 2950 PSI). The unit was operated for 1.5 minutes at 3500 PSI and 15,000 RPM. After proof pressure test, the overall efficiency was 71.5% at 2950 PSI. The overall efficiency was 72.3% at 3500 PSI. The volumetric efficiency was 94.7% at 2950 PSI before and after the proof pressure test.

Sampling Test

The unit was cycled 31,588 times to complete the sampling test. The total test time for the sampling test was 70 hours. The patch test at the completion of the sampling test (See Figure 20) indicated the presence of metal. Examination of the unit, after disassembly, showed the front radial bearing had worn on the rollers and races.

This bearing had been contaminated by two previous unit failures. In addition, the bearing was inadvertently left in the housing during the rework machining. The failure was not attributed to a deficiency in design. The bearing was replaced and testing was continued.

Low Temperature Test

After soaking 20 hours at -75°F., the pump was started and brought to rated speed with minimum back pressure. The maximum torque during acceleration was 50-pound inches. The test was repeated twenty (20) times.

Five starts were made with the load valve set at 2800 PSI (full flow). The unit temperature was stabilized at $-60 \pm 5^\circ\text{F.}$ before each start. The unit was accelerated

8-0108-078-310-304
FUNCTIONAL TEST
PATCH CHECK

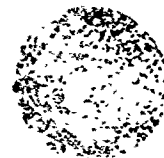
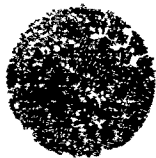
INLET

OUTLET

CASE



PRIOR TO TEST



AFTER TEST

Figure 20 - Patch Samples

Low Temperature Test (Continued)

to rated speed and operation maintained for two minutes. The maximum torque recorded was 120 pound-inches.

Five starts were made with the load valve completely closed (zero flow). The unit temperature was stabilized at $-60 \pm 5^{\circ}\text{F}$. before each start. The unit was then accelerated to rated speed and operation maintained for two minutes. The maximum torque recorded was 83 pound-inches.

Calibration After Cold Starts

After the low temperature tests, the unit was calibrated. Overall efficiency at 15,000 RPM and 2900 PSI was 71.9%. During calibration, the unit failed (total running time since beginning 189 hours). The failure was caused by cracked piston rollers. The rollers were redesigned (See comment below).

It was also noted during disassembly that the Teflon kidney seals were fretting. The fretting was caused by the Teflon cold flowing through the .004" rotating group to housing clearance.

Modification of Unit

The unit was modified incorporating a redesigned piston roller. After two hours of operation, a malfunction occurred. The redesigned rollers did not contribute to the failure. The rollers were in "as new" condition. The failure was caused by silver plate wear and contaminants in the annulus groove reducing the piston clearance. The reduced clearance caused the piston to score the drive plate and resulted in the unit having excessive internal leakage.

Examination of the pistons drive plate and annulus plates showed damage to only one set of pistons and one annulus plate. The drive plate on the side next to the damaged annulus plate showed heavy wear from the pistons. This indicated an unbalance or that contamination shifted the piston toward the annulus plate. The drive plate, piston and annulus plate on the opposite side was undamaged.

The clearance between the piston sleeve and drive plate slot were within print limits (See Table III).

The tests were stopped at this time. The total accumulated time on the unit was 196 hours.

SUMMARY OF TEST RESULTS

1. The development tests on the orbital pump have proven that the pump concept has the potential to achieve the 500 hour endurance life.
2. Pump problems encountered during the development test indicate further investigation is necessary on the pump rotating group.
3. This basic design using a cartridge concept provides ease of maintenance. This is particularly advantageous for field repair.
4. The overall efficiency of the pump is not as high as anticipated.
5. Pressure pulsations and noise level of the pump are excellent.

The following is a tabulation of the 196 hours of pump operation and the test incidences involved.

TOTAL HOURS

TEST DESCRIPTION

57	After completing the Break-In, Performance, Horsepower Loss, Functional and Calibration tests, a piston bound in the annulus groove while operating at 19,800 rpm.
74	The unit was rebuilt with increased piston clearance and had a similar failure during the initial Performance Tests.
189	The unit was then rebuilt with additional piston clearance, reduced silver plating thickness and changes to the porting and hydraulic balance. The pump then completed the Break-In, Performance, Proof Pressure and Sampling tests. At a normal inspection, the front radial bearing rollers and races were found to have some wear. The bearing was then replaced and testing continued.
196	The low temperature test and calibration tests were then completed. It was then found that the unit had cracked piston rollers. Modified rollers were incorporated and tests continued for an additional 2 hours.

TABLE III

PISON CLEARANCE IN INCHES

(After Failure)

	<u>DRIVE PLATE</u>	<u>SLEEVE</u>	<u>CLEARANCE</u>
(1)	.3500	.3502	.0002
(2)	.3500	.3502	.0002
(3)	.3500	.3502	.0002
(4)	.3500	.3502	.0002
(5)	.3500	.3502	.0002

SECTION VI

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

- I. The design investigation showed the orbital pump design incorporated the best features for a high speed engine driven pump.
- II. The pumping concept was proven as a result of the 196 hours of endurance operation.
- III. The noise level of the orbital pump appears to be lower than other type pumps when operated under comparable operating conditions.
- IV. Further development of materials and machining techniques are necessary for extended life.

RECOMMENDATIONS:

1. As a result of these investigations, further development of the orbital pump is recommended for increased reliability and performance. This future development would include the following:
 - a. Initiate a material investigation for those parts that require further development.
 - b. Perform evaluation testing of the recommended material and configuration.
 - c. Endurance test of final configuration selected for a minimum of 500 hours.
 - d. Qualification test.
2. The present pump design would satisfy requirements for limited life aerospace applications.

Unclassified
Security Classification

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4. DESCRIPTIVE NOTES (Type of report and inclusive dates) Final Report 1 June 1964 to 1 October 1966		
5. AUTHOR(S) (Last name, first name, initials) Perian, Frederick W.		
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c. Task No. 812807	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report) None	
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11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY Air Force Aero Propulsion Laboratory Research and Technology Division, Air Force Systems Command, Wright-Patterson Air Force Base Ohio 45433	
13. ABSTRACT This report covers an exploratory development program to establish design technology for hydraulic pumps capable of being driven directly from aircraft gas turbine propulsion engines. The program was divided into a study phase, design phase, fabrication phase, and a test phase. After completing the study phase, the orbital type pump was chosen by the Air Force Propulsion Laboratory for design, fabrication, and test. Other pumping concepts were available that would theoretically reach the contract goals. However, it was desired that a unit be developed that would show growth potential beyond 10 gpm and 18,000rpm. The .140 cubic inch per revolution orbital pump was designed from an existing .70 cubic inch per revolution unit by using a .585 scale factor. The porting was designed for 10 gpm flow, and the bearings were designed for 1000 hours life. The pump was designed to be compatible with MIL-H-5606 fluid in a Type II system. The test phase was conducted in accordance with the development test program for the E80108075 orbital piston pump. The tests were to determine pump performance and endurance capabilities. An investigation was also made to determine the horsepower loss for the various rotating parts of the pump. The orbital pump completed the Examination of Product, Break-in Run, Functional Test, Calibration, Proof Pressure Test, Sampling Test, and Low Temperature Test.		

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Unclassified
Security Classification

14 KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Pump						
Variable						
Piston						
Orbital						
Hydraulic						
High Speed						
Engine Driven						

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